AUTOMOBILE ENGINEER

PRODUCTION MATERIALS

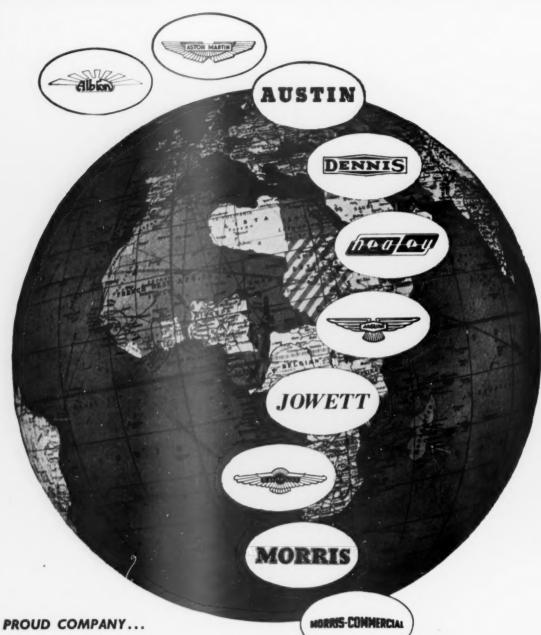
Vol. 43 No. 566

MAY, 1953

PRICE: 3s. 6d.



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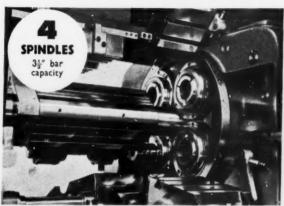


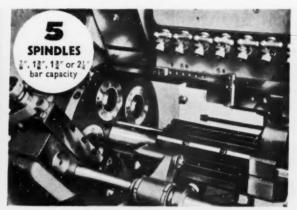
For complete reliability you can depend on

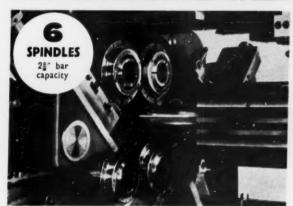
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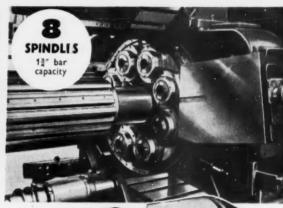
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Wickman 5-spindle automatics are also available for chucking work in 5" and 6" capacities.

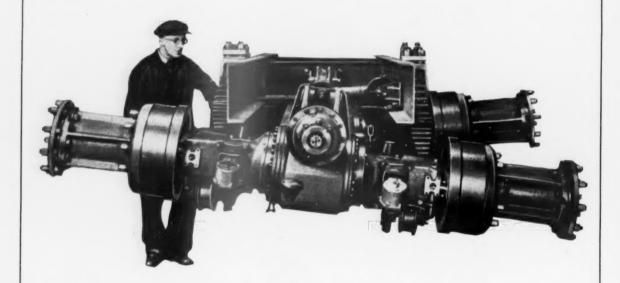
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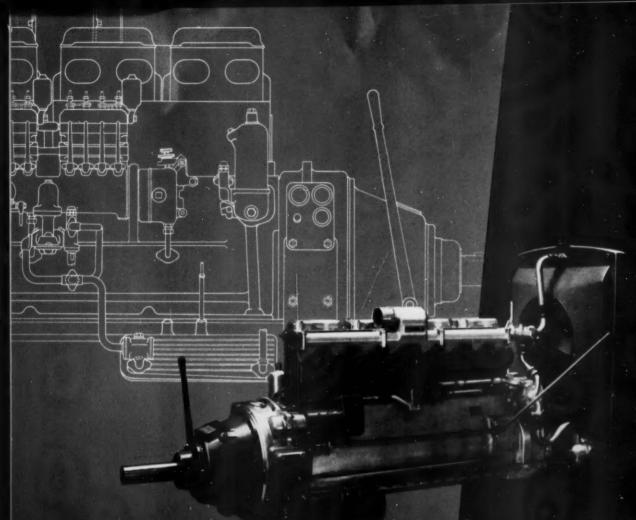
timum service.

The illustration shows a Blanking Die and Taps Blanking Die ither Firth made from either Brown "NONVAR" or Brown "NONVAR" tool and die steels.

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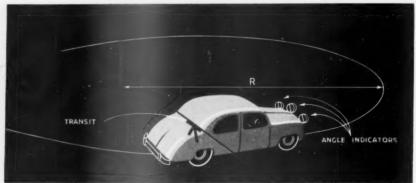
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This sketch of a car on a steering pad shows the dial indicators for front wheel angles. The transit is used to measure roll and rear drift angles.

1: Causes of over- and under-steer. Introduction

ALTHOUGH the terms over-steer and under-steer were unknown twenty years ago, most people today have a fairly good idea of what they imply. Many of us were familiar with their cruder manifestations, such as the uncontrollable swaying of the back of the car when a sack of potatoes was carried on the luggage grid, or

the car which refused to corner fast no matter how much the steering wheel was lugged round; most of us equally had no idea of what was really happening and what caused it all.

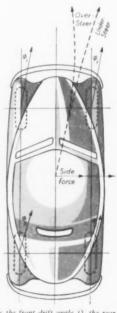
The first published information about the characteristics of the pneumatic tyre which led to such behaviour appeared in the S.A.E. Journal for February 1935, when the famous paper by R. D. Evans was printed, and the latest version of the everlasting triangle was presented to us-the struggle between 'enveloping power', cornering power and durability. A good deal more is known about tyres today than then, but although sizes, rim widths and inflation pressures have all changed, the fundamentals (like this particular version of the triangle) are unaltered.

But the tyre behaviour is only half the story, and it was some time further still before all or even most of the implications of tyre behaviour, as affected by the car properties, were grasped at all

widely. Roll centres, roll axis, roll stiffness, centreof-gravity height and fore-and-aft position all affect the angle of roll of the car for a given degree of cornering, and the angle of roll is only one of the many characteristics which determine whether under- or over-steer will be present, and in what degree. The other

> major characteristics are the geometry of the front and rear suspensions, affecting the attitude of the different wheels in relation to the road, and the weight transference at both ends of the car caused by the centripetal accelera-

It may be as well here to clarify the method of denoting the severity of a turn which is being negotiated. This is now universally known by the proportion of g, the acceleration due to gravity, which this centripetal acceleration assumes. For instance, if v be the vehicle speed in ft./sec. and r the radius of the turn in feet, then the centripetal acceleration is well known to be f ft./ \sec^2 and if $\frac{v^3}{rq}$ is say, 0.5, then the sideways acceleration, or severity of the turn is 0.5 g. Using a radius of turn of 108 ft. which is the largest standard on the steering pad on the M.I.R.A. Proving Ground at Lindley, then for a 0.5g turn $v^{1}=0.5 \times 32.2 \times 10^{-2}$ 108 = 1738.8 and v is 41.7 ft/sec. or 28.4 m.p.h.



O, is the front drift angle, Oz the rear. If O_1 is greater than O_2 , the centre of the circle about which the car is turning will lie to the right and the car will understeer. If O_1 is less than O_2 , the

AUTOMOTIVE PRODUCTS COMPANY LTD., LEAMINGTON SPA, ENGLAND

Thompson Self-adjusting

STEERING ROD ASSEMBLY



keeping busy?

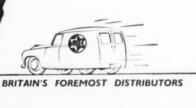
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"I know, it's a bit like that, these days. Still, some things are easier . . . twist drills, for instance."

"Oh! . . . twist drills . . . I never bother to line them up at all — Monks & Crane carry such phenomenal stocks."

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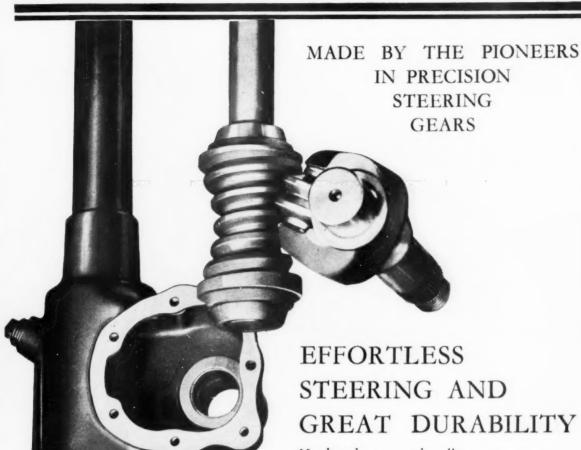
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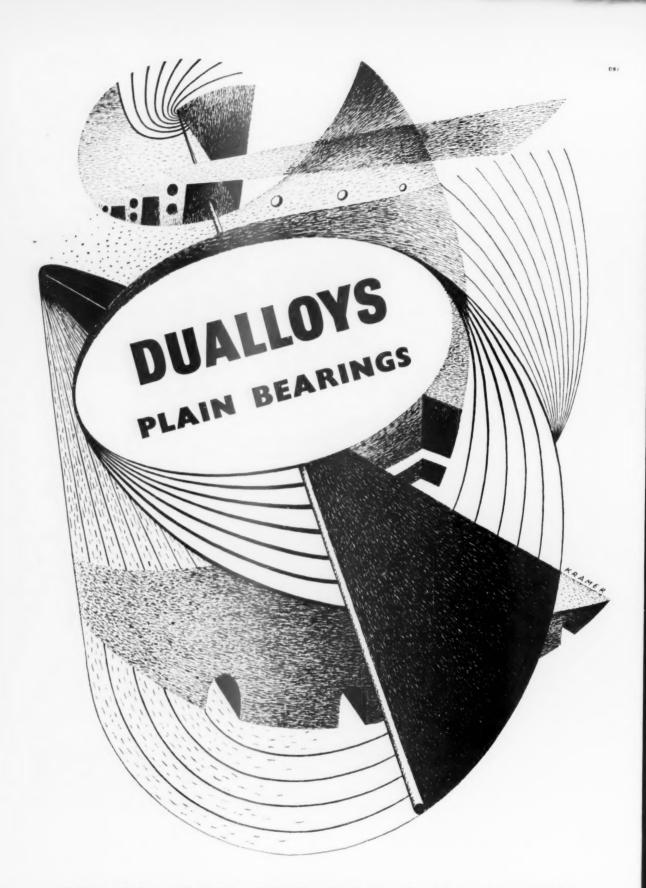
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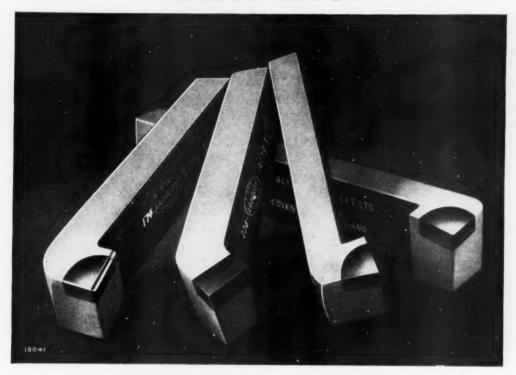
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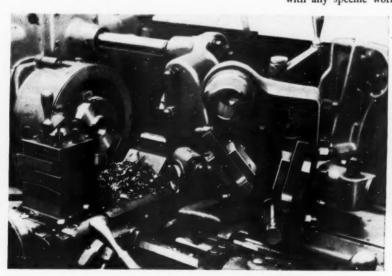
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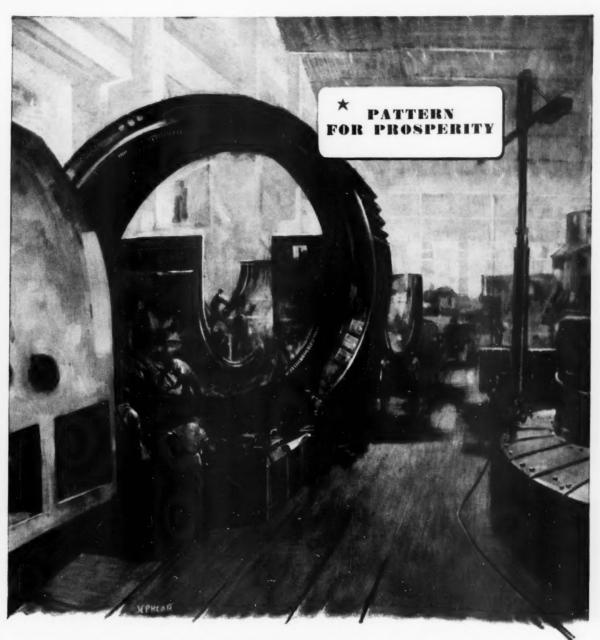
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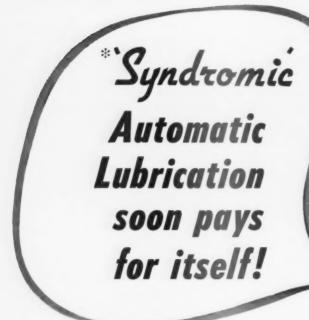
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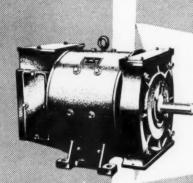
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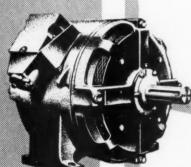
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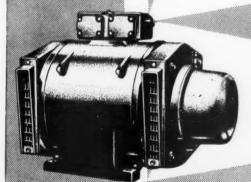


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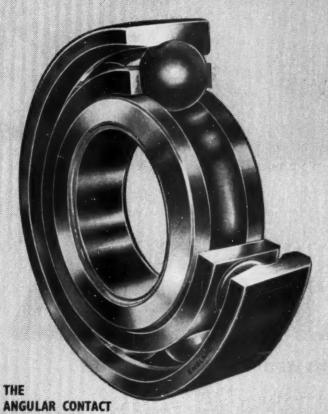




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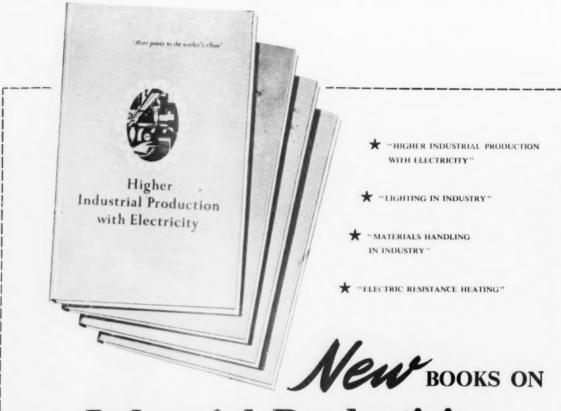


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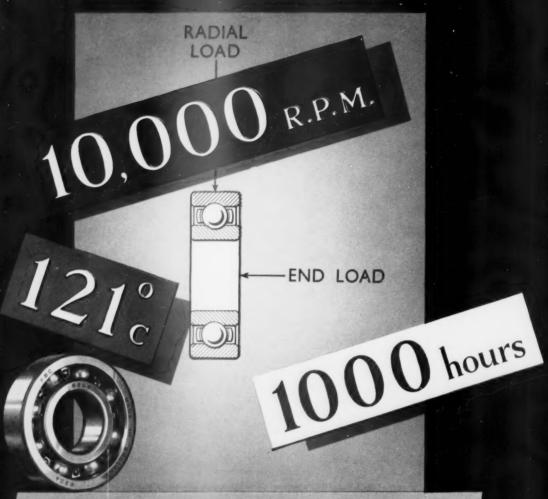
Our colour photograph shows Hub caps by courtesy of the Ford Motor Company. Steering Wheel by courtesy of Wilmot Breeden Limited.



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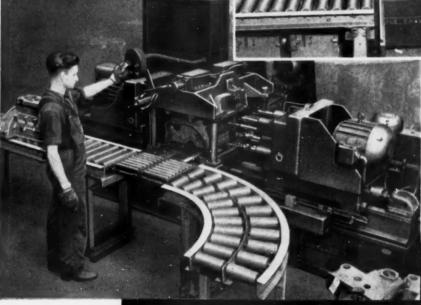
ARCHDALE specials are specially designed for high production at low cost. They play a big part in the production of the new FORDSON MAJOR tractor and the illustration opposite shows a 3 way multi-drill engaged on cylinder blocks. Opera-

The lower illustration shows a second machine drilling the valve and water jacket sides of the block and the auxiliary drive bore.

drilled.

and 1- \(\frac{1}{16}\)" holes drilled and 3-\(\frac{1}{16}\)" holes drilled and 3-\(\frac{1}{16}\)" holes drilled and 10-\(\frac{1}{2}\)" holes





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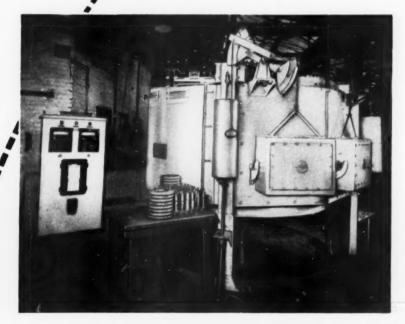
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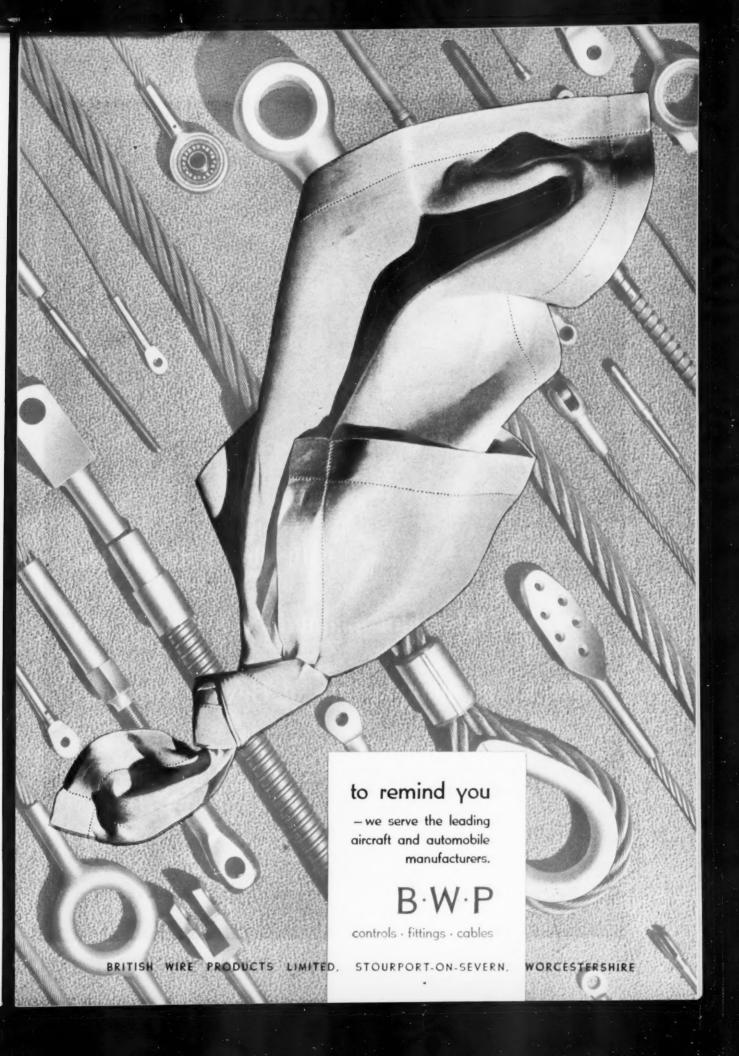


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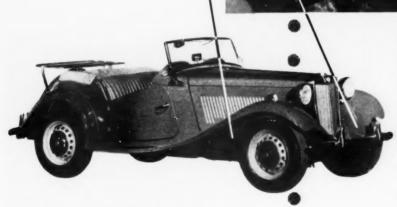
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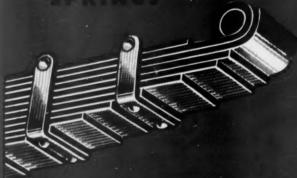
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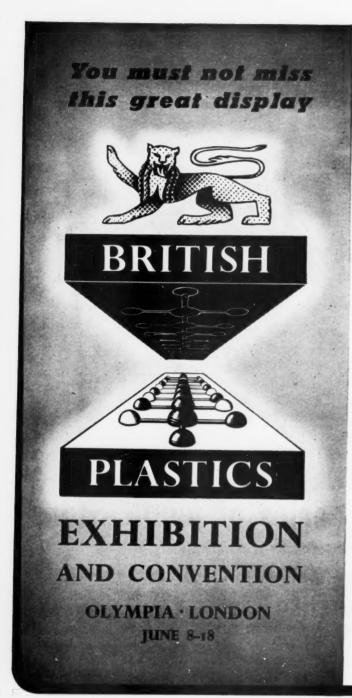
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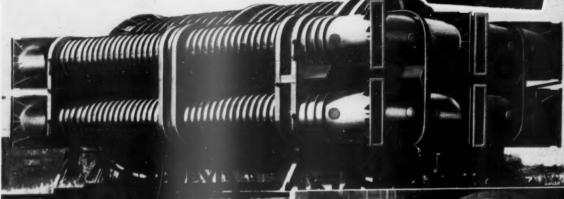
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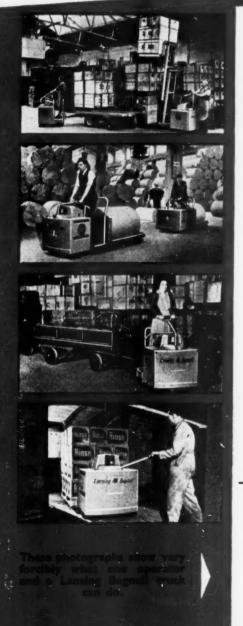
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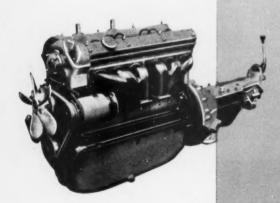
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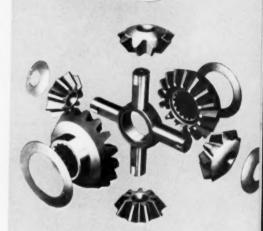
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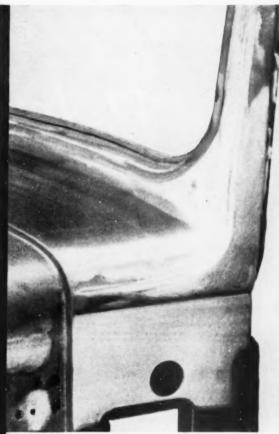
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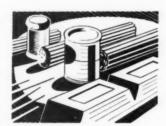
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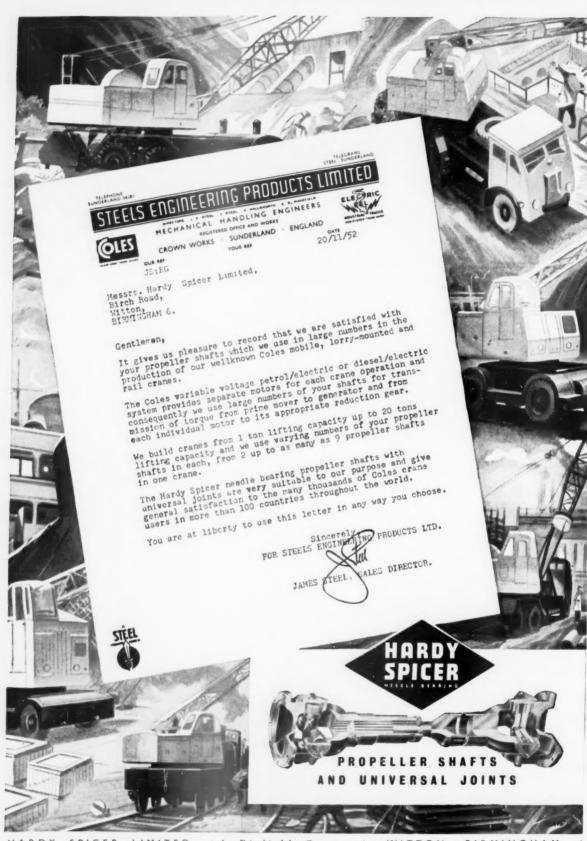
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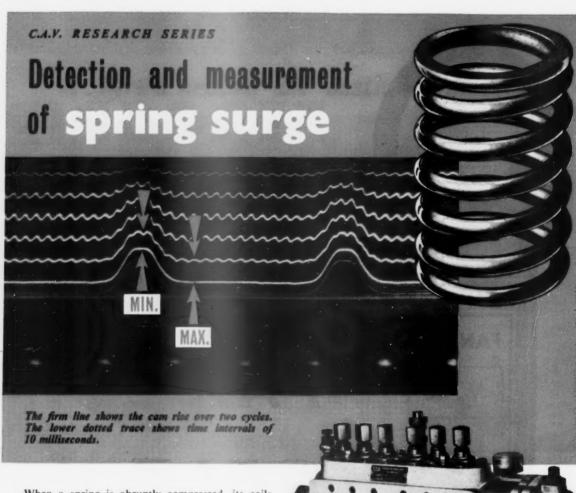
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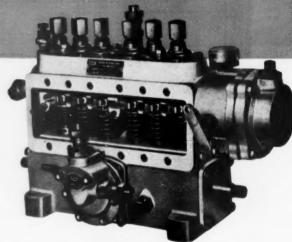
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The illustration above is from a research laboratory record of surge in a spring, the wavy lines showing the displacements of individual coils. The stress range in any turn can be deduced by observing the minimum and maximum turn-to-turn distance.



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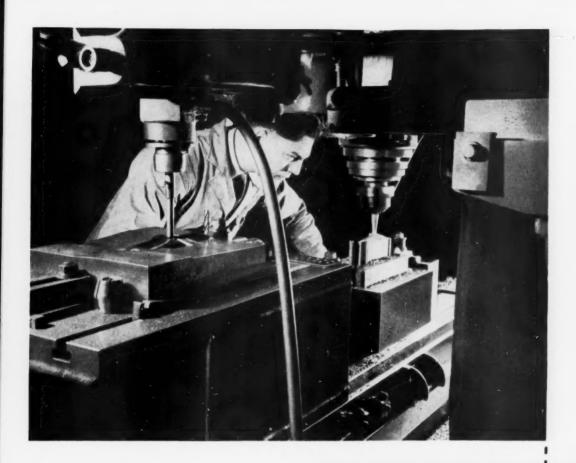
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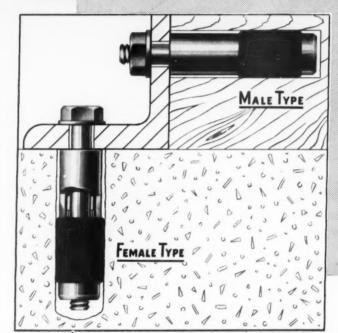


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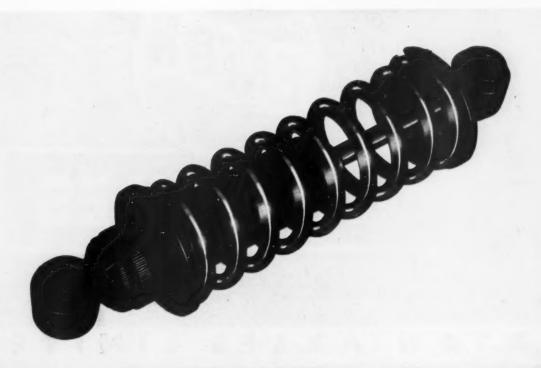
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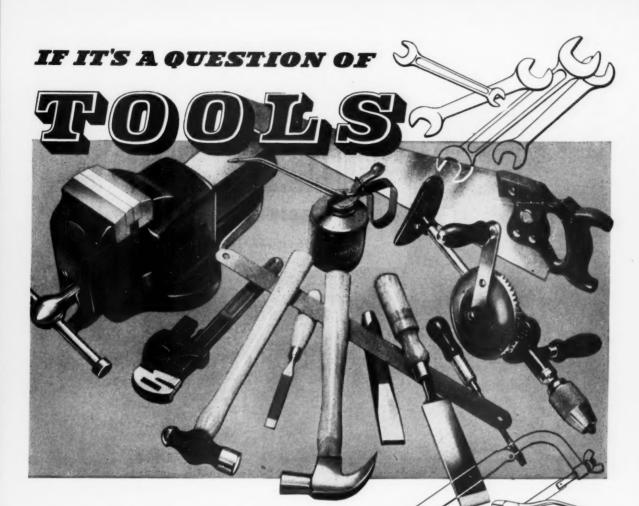
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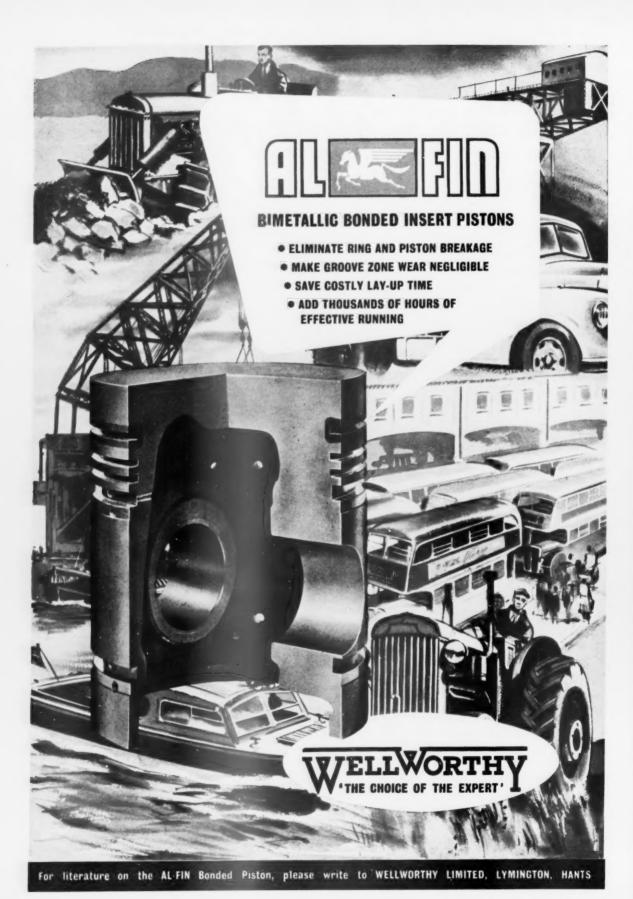
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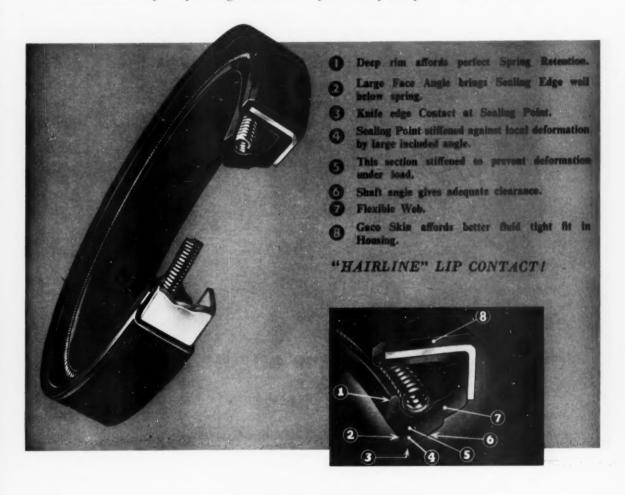
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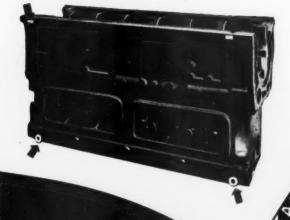
More miles per shilling! That's what BP Super gives a car.

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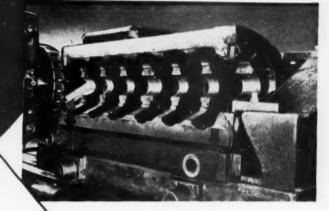


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Milling this Jeyland casting CANT



Leyland "600" cylinder-block castings (top left) supplied by the West Yorkshire Foundry are furnished with two machined faces and two spot facings and dowel holes, as illustrated. The other photos show a casting set up for initial milling operations in the Leyland machine shop. The top picture clearly shows the dowelled locations, and the bottom one indicates where the other two locating

surfaces make contact with the fixture. There's nothing to go wrong; each casting has already been checked to machined locations AT THE FOUNDRY and cannot fail to clean up on all machined surfaces. Every West Yorkshire Foundry job undergoes the same rigorous inspection and is delivered ready to work on. And, remember, our modern production methods bring down costs to the minimum.

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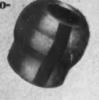


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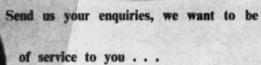


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Design, Materials, Production Methods, and Works Equipment

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Two-stroke Engines

TREND that has been most marked in post-war Continental motor shows is the growing use of small two-stroke engines in private cars. This has undoubtedly arisen primarily as a result of economic conditions which have made the very light austerity car so popular in European countries. The two-stroke engine, because of its relative simplicity and low cost, is well suited to the small car, and Auto-Union D.K.W., Champion, Goliath, Gutbrod, Hanomag, Lloyd and Vidal have all competed in this field.

None of the large British manufacturers has so far produced an economy car. However, contrary to popular belief, the subject has been given careful and detailed consideration. Indeed, some designs of this type have got at least as far as the drawing board stage, and have then been rejected. There are a number of reasons why manufacturers have not proceeded further with these projects.

One of the reasons is that British manufacturers have concentrated on production for dollar markets where this type of car is not in demand, and the available productive capacity has been devoted to the classes of vehicle that will sell in North and South America. It does not necessarily follow that these conditions will continue.

Advances in design

The considerable advances that have been made in twostroke engine design have gone a long way towards offsetting the disadvantages of that type of engine. At one time these power units had a relatively low specific output in terms of b.h.p./litre, they were noisy, brake specific fuel consumption was appreciably higher than that of comparable overhead-valve four-stroke engines, and there was a marked tendency towards four-stroking at idling speeds and under light load.

In modern designs incorporating the crankcase compression system, the specific output is about 35 b.h.p. litre, which compares favourably with present-day overhead-valve four-stroke engines. This result is obtained with the Schnuerle scavenge system, the basic patents for which lapsed in 1949. It obviates the need for deflector pistons, thereby making possible the incorporation of hemispherical combustion chambers and the use of higher compression

Another benefit derived from flat or slightly domed piston crowns is freedom from thermal distortion of the ring belt, and this makes it possible to allow smaller clearances between the pistons and cylinder. The smaller clearances lead to quieter running, but the noise heard as the ports are uncovered still remains. However, modern intake and exhaust silencers are capable of eliminating most of this.

The Schnuerle system has reduced brake specific fuel consumption by as much as 25 per cent as compared with that of some of the earlier engines. Four-stroking is perhaps not so serious a defect in four as in single cylinder units. Apparently it can be markedly reduced by the employment of a matched silencing system. The tendency can also largely be eliminated by petrol injection. This is expensive, but the cost can be appreciably reduced by production on a large scale, partly because development charges represent a fairly high proportion of the total cost.

The use of aluminium for cylinder heads presents few problems with the two-stroke engine because there are no valves and seats to accommodate. With heads of this material, not only is cooling improved so that still higher compression ratios can be used, but appreciable weight saving is effected. In small cars where the power-to-weight ratio is critical, this weight saving is important. It is even more beneficial in rear engined vehicles, which normally tend to have too much weight at the back.

Engine layout

The simplicity of induction and exhaust obviously leads to a saving in prime cost, but equally important is the elimination of maintenance work to rectify sticking and burned valves, worn guides, tappet adjustment, etc. Because there is no valve gear, the overall height of the engine is reduced, a most desirable feature so far as installation in very small cars is concerned.

Employment of crankcase compression means that the choice of engine layout is not so wide as with four-stroke engines. Flat four and flat twin arrangements with crankcase compression engines create certain difficulties, and blown units are out of the question for economy cars. Single cylinder engines are generally too small and they suffer from the same disadvantages so far as out-of-balance forces are concerned, as four-stroke engines. The out-of-balance couples in vertical twin cylinder layouts are not easily isolated from vehicle structures. However, a conventional four-cylinder two-stroke arrangement is appreciably cheaper and lighter than a similar four-stroke unit.

There are good reasons for seriously considering the two-stroke engine for the economy car. Because of the absence of valve gear, this type of unit could be made at a very low cost, particularly if produced in large quantities.

Further manufacturing economies result from the use of the petroil lubrication system, although oil consumption is heavier. It is true that because of lack of previous experience with this class of engine, the development period would probably be a long one.

Rejected Work

ARADOXICAL though it may seem, it is not generally sound policy to organize large quantity production to give 100 per cent acceptable work. Invariably, the governing factor should be the lowest possible unit cost for work to a specified standard and at first sight it may appear that this condition would best be met by a system that produces 100 per cent acceptable work. But closer examination of the question will generally show that the maximum economy is attained when the manufacturing processes are such that a small percentage of the articles produced fall outside the specified standard.

It is almost unnecessary to say that as far as possible the product supplied to the user should be in accordance with specification, but that is not to say that it is either possible, or desirable, completely to eliminate manufacturing rejections. Practically every machine and every process is subject to random variations in some degree, and attempts completely to overcome the results of these variations can only lead to increased manufacturing costs, particularly where the tolerances are close.

Unfortunately, seldom is trouble taken to determine the percentage of faulty work that may be considered permissible, and there is no doubt that in many factories production superintendents and foremen must try to attain an impossibly high standard, with a consequent adverse effect on costs. In addition, where the percentage of rejected work has to be kept below an unrealistic figure, work is frequently re-conditioned that it would really be cheaper to scrap out of hand.

So many factors affect the economics of permissible scrap percentage that it is impossible to lay down any rule that can be applied over even a small range of articles, and a general rule is manifestly impossible. The economics of scrap percentages is a subject for empirical investigations. Such investigations could well lead to salutary lessons for production engineers; they could also be of value in determining whether the design tolerances are realistic.

It must be made clear that this is not a plea for a reduction in standards of quality. On the contrary, a more realistic approach to the question of rejected work could be the means of improvement in quality. For example, in some cases it might well disclose that closer tolerances could be specified without increasing the rejections to an uneconomic figure. Furthermore, a realistic appraisal of the permissible percentage of faulty work would lessen the temptation to keep the rejections to an artificially low figure by accepting work below the specified standard. This is a temptation that most production and inspection executives meet from time to time.

Power Output

URING recent months several American automobile manufacturers have announced new passenger car engines with maximum power output exceeding 200 h.p. No sooner did one manufacturer introduce an engine of very high maximum power than others followed with even higher power engines. Obviously there is very keen competition in this matter but it is difficult to see what good purpose will be served by these developments. Even many Americans are wondering whether they are worth while. In circles closely connected with the American automobile industry, there is a feeling that what happened with diesel engines in 1936 is now to be repeated with passenger car engines. About the middle 1930's there were similar competitive increases in power output from diesel engines. When the duty for which the diesel engines were to be used is considered, there may have been justification for increasing the power output, but it should be remembered that the increases led to a host of crankshaft and bearing troubles.

It does appear that a high maximum power output is being used solely as a selling point and we must take leave to doubt whether it really represents a worthwhile advance. The information available at the moment refers only to maximum output; what happens at the lower end of the power curve is a secret known only to the engine makers. However, it is highly probable that the very high maximum has been obtained at the expense of the output over the range of speeds that will be generally used. The high maximum output will also probably entail an engine with increased petrol consumption. This may be a minor point in the United States of America but it is a major one in this country and in all other important world markets. In our opinion, this craze for normal production engines capable of producing more than 200 h.p. may well be left to the

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ALLARD PALM BEACH CHASSIS

A Light Car with a Tubular Frame, and Powered Either by the Ford Zephyr or Consul Engines

PRICE has now become a decisive factor in selling cars both at home and overseas. Because of this, the Allard Motor Co., Ltd., of Clapham, S.W.4, have designed the Palm Beach model, which is the smallest of their post war vehicles. The 21C model is powered by the Consul engine, and its dry weight is 1700 lb, while with the Zephyr installation, the 21Z model, the weight is 1780 lb. With the smaller power unit, the front/ rear weight distribution is 55:45, and with the larger one, 57½:42½. This should tend to give a slight understeering ten-

dency and help to give good handling characteristics. The wheelbase is 8 ft, and the front and rear track are respectively 4 ft 3 in and 4ft 1 in. With the hood up, the overall height is 4ft 3 in. The other principal dimensions are: width 4ft 10 in, length 13 ft, ground clearance 5 in to the exhaust silencer and 7 in to the frame. It is stated by the manufacturers that frontal area is about 10 ft2.

SPECIFICATION

CONSUL ENGINE: Four cylinders. Bore and stroke $3\frac{1}{8}$ in $(79\cdot 4 \text{ mm}) = 3 \text{ in}$ $(76\cdot 2 \text{ mm})$. Swept volume 1,508 cm³. Maximum b.h.p. 47 at 4,400 r.p.m. Maximum b.m.e.p. and torque respectively 121 lb in² and 74 lb-ft at 2,400 r.p.m. Compression ratio 6-8:1. Three-bearing, cast steel, fully balanced crankshaft. Push rod operated overhead Zenith downdraught carbu-

rettor, with a 26 mm choke. ZEPHYR ENGINE: Six cylinders. Bore and stroke $3\frac{1}{8}$ in $(79\cdot4\text{ mm})\times3$ in $(76\cdot2\text{ mm})$. Swept volume $2,662\text{ cm}^3$. Maximum b.h.p. 68 at 4,000 r.p.m.Maximum b.m.e.p. and torque respectively 122 lb.in² and 112 lb-ft at 2,000 r.p.m. Compression ratio 6.8:1. Four-bearing, cast steel, fully balanced crankshaft. Push rod operated overhead Zenith downdraught carbuvalves. Zenith downdrau rettor, with a 30 mm choke.

CLUTCH: Ford single dry plate, 8 in diameter, with hydraulic withdrawal

GEARBOX: Three forward speeds and one reverse, baulked synchromesh on top and second. Ratios: top 1:1, second

1-642:1, first 2-48:1, reverse 3-86:1. In later models they will be: top 1:1, second 1:692:1, first 3:273:1, reverse 3:975:1.

REAR AXLE: Ford Zephyr. Hypoid. 3-floating with banjo casing. by open propeller shaft. Ratio 4-44:1. SUSPENSION: Rear, coil springs with almost parallel trailing links. Armstrong AT7 telescopic shock absorbers, in bore 7 in stroke. Front, divided axle and coil springs. Armstrong AT7 telescopic shock absorbers, & in bore < 4 in stroke. STEERING: Adamant Marles Hourglass worm and roller. Ratio 14:1. 2½ turns from lock to lock. Turning circle 28 ft. BRAKES: Girling hydraulic, 2LS front. Drum diameter 9 in. Shoe width 1½ in.

Friction lining area 121 in².

TYRES: 6·40×13·0 at 18 lb in² front and rear, or 5·50×15·0 at 20 lb in² front and rear.

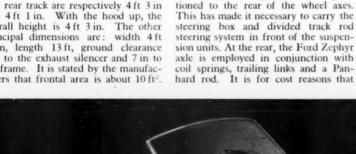
DIMENSIONS: Wheelbase 8 ft. Track, Front 4 ft 3 in, rear 4 ft 1 in. Ground clearance minimum 5 in. Overall length 13 ft. Overall width 4 ft 10 in. Overall height 4 ft 3 in with hood up. Frontal area 10 ft². Dry weight: Consul 1,700 lb, Zephyr 1,780 lb.

this arrangement has been adopted in preference to the De Dion layout used in previous models. Except for welding on one bracket for the Panhard rod and lugs for the upper trailing links, the axle unit is used as received from Fords.

A tubular frame has been adopted because it is simple to repair, and modifications to the design may be more easily made if required at a later date. No pressings are employed, and even circular dished components, such as suspension spring pans, are fabricated by welding. Two petrol tanks are employed, one over each rear

wheelarch, and a filler tube projects through each rear quarter. The advantage of this arrangement is that it has made possible the provision of an exceptionally large boot. The spare wheel is carried on the boot floor.

One of the greatest difficulties that the small motor car manufacturer has to face is that of providing an adequate spare parts service. This is particularly

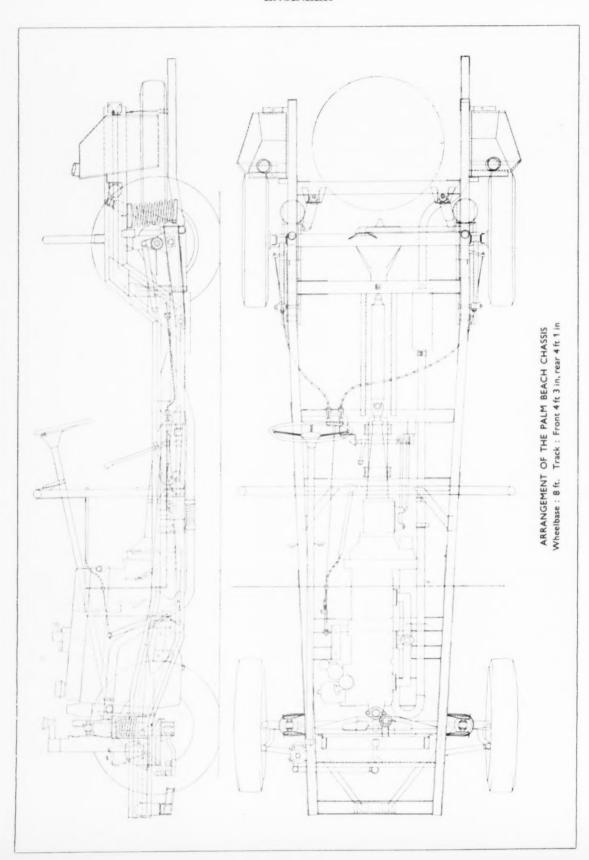




In order to accommodate the divided

front axle, the engine has been posi-

In the Allard Palm Beach, the petrol tanks behind the rear wheel-arches do not restrict luggage space



true when, as in the case of the Allard Motor Co, a large proportion of the vehicle output is sent overseas. As with their earlier models, the Company have solved the problem by employing Ford mechanical components, and where it has not been possible to use complete Ford assemblies, as for instance in the front wheel hubs, nearly all the parts subject to wear are identical with those used in the Ford units.

Engines

The Consul and Zephyr engines were described in the September 1951 issue of the Automobile Engineer. With the Consul engine, which develops 47 b.h.p., the power to weight ratio is 62 b.h.p./ton, and with the Zephyr engine the ratio is 85·5 b.h.p./ ton. This should be adequate, even with the smaller power unit, to give the vehicle a lively performance.

Unfortunately, the swept volume of the Consul engine is 8 cc over 1) litres, and this places it at a serious disadvantage from the point of view of competition enthusiasts. arrangements have been made with Burtonwood Engineering Co., Ltd., of London, N.W.9, to fit sleeves and reduce the volume to about 1,495 cc. At the same time, this Company, in Allards, conjunction with developed a different inlet and exhaust manifold arrangement to take two Zenith carburettors and to incorporate exhaust heated hot spots.

In place of the exhaust pipe bolted to the head in the Ford layout, four separate cast iron elbows, with their joint faces appropriately profiled, are bolted on. The elbows are directed downwards and connected by flexible pipes to branch pipes on the exhaust. On top of each of the four elbows is a port round which is a flange forming a joint face. Bolted to the joint faces on each adjacent pair of elbows is a flanged interconnecting pipe of cast aluminium. These pipes both have a port above their centres where they are machined and flanged for the bolts securing them to the inlet elbows below which they form a hot spot. Two

cast aluminium inlet elbows a r e employed on this engine. Each carries a Zenith downdraught carburettor on top, and has its lower end bolted to the cylinder head to serve two ports.

This installation, although more expensive, is more efficient than the standard layout, and the power developed by the engine is considerably increased. Further improve-

ments may be obtained by using higher compression pistons. However, figures cannot yet be quoted because the tests have not been completed by the time of going to press. The new arrangement, as an alternative to the standard one, will shortly be available on the Consul engined cars, and on the Zephyr it will be possible, by using an extra set of the same components, to fit three carburettors. In all cases it has been necessary, because of the improved power output and higher r.p.m. to fit Vandervell lead bronze big end bearings in place of those used in the normal production engines.

Transmission

With all power units, the Zephyr gearbox is used. The ratios are given in the specification panel. A Hardy Spicer open propeller shaft transmits the drive to the rear axle. It is 2 in diameter by 31 in long in the Consulengined installation, and 223 in long in the Zephyr. Needle roller universal joints are employed at both ends and the sliding joint is carried in the rear extension of the gearbox. Provision is now being made to fit an overdrive The installation of this will involve lowering the centre of one of the frame cross members about 11 in. A modified Ford steering column

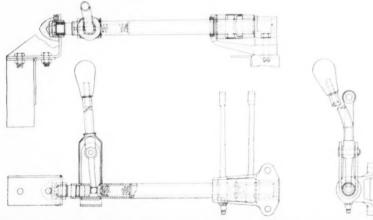
The twin carburettor arrangement on the Consul engine

gear shift mechanism is mounted under the body floor. The lever is more or less vertical, and offset from the longitudinal centre line of the car about 3 in towards the driver. It is claimed that this arrangement gives a more positive gear shift motion than is possible with the unit mounted on the steering column. This advantage is, of course, offset slightly by the obstruction on the floor, but those who buy a car of this type prefer the floor mounted There are two more advantages in using the Ford mechanism. One is that the mechanism was originally designed with a motion appropriate to this gearbox, and it is therefore relatively easy to adapt it. The second advantage is that replacements for parts subject to wear are readily available.

The Ford steering column control tube, which is \$\frac{1}{2}\$ in diameter, is shortened and positioned laterally under the gearbox rear extension. It is carried by two brackets, one at each end, welded to a frame cross member. Brazed inside the tube about $2\frac{1}{6}$ in from one end is a solid steel insert, and a peg is passed diametrically through both the tube and the insert in such a manner that each end projects about \$\frac{1}{6}\$ in. The adjacent end of the tube is carried in an aluminium die-casting bolted to the bracket on the cross member; its bearing length in the die-

casting is about 11 in.

Integral with the die-casting is an arm extending about 21 in towards the centre line of the vehicle. The end of the arm, on which a boss is formed, is turned forwards to form, with the tube bearing, an offset fork between the two prongs of which the operating lever bosses are a running fit on the tube. In the arm, a grease seal is pressed into a recess in the face remote from the lever bosses. On each side of the tube and insert, the peg ends project into D-slots in the inner When the ends of each lever boss. levers are in the neutral position, all four of these slots are in line to form two diametrically opposed long slots. To select a gear the tube is slid axially until the peg ends engage in one pair of the slots, and it is then rotated to the operating lever either forwards or backwards according to



A modified Ford gear shift mechanism is mounted transversely on the chassis frame in the Palm Beach



In the Palm Beach, twin trailing links are employed in conjunction with coil springs and the Ford Zephyr rear axle

which gear is to be selected. The link between the end of each lever and the gearbox is a simple tube with rod inserts in its ends. These inserts are bent at right angles to engage in the holes in the ends of the levers where they are retained by split pins and plain washers. At the control end the insert is welded in, but at the gearbox it is screwed into the tube, and secured with a lock nut.

A fabricated steel bracket, bolted to the bracket on the frame cross member, carries a steel insert in the other end of the control tube where the gear shift lever is mounted. This insert is a running fit in the tube, and its outer end is shouldered and bears in a bush. The bush is of an absorbent material to retain the lubricant, and is in a steel channel section ring which is pressed into a tubular housing welded in a bole.

in the support bracket. Axial location is effected by a horseshoe washer engaging in two diametrically opposite grooves on the insert. This washer is retained in position by a domed cap passed over the end of the insert and carried on the tubular housing for the bearing ring. A grub screw in a hole in the side of the cap engages in the housing to lock the assembly.

The gear shift lever is pinned between two lugs welded to the control tube. Its lower, or striker end projects through a hole in the tube into a large notch in the steel insert. A rubber gaiter

protects this part of the assembly from dust. The pivot pin is positioned with its axis fore and aft so that when the lever is moved from left to right the lower end is held stationary in the notch while the lugs and tube slide from left to right with the lever. If a fore and aft motion is given to the lever, the whole assembly comprising the tube, insert, and an operating lever at the other end rotates about the axis of the tube.

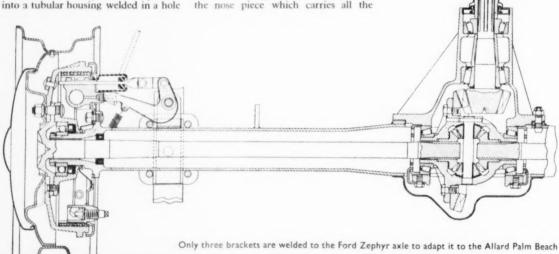
The Zephyr-Six rear axle is fitted on both the 21C and 21Z models. It is a hypoid bevel, three-quarter floating unit with a banjo type casing. The final drive ratio is 4-444:1. Hypoid gear oil is recommended and the capacity is 2½ pints.

Black heart malleable cast iron, B.S. 310: 1947 grade 2 is employed for the nose piece which carries all the

gears, and which is secured to the casing by eight & in diameter bolts. The axis of the En 35A hypoid pinion is offset 0.88 in to the right of the plane of the differential spindle axis, and 1.375 in below the crown wheel axis. Two taper roller bearings spaced approximately 3 in apart, between centres, support the En 35A crown wheel pinion spindle which is over-hung in the usual manner. The 9-toothed pinion is upset forged on the end of the spindle. At the front bearing, the spindle diameter is 1 in and at the rear one it is 11 in. Splines on the front end of the spindle transmit the drive from the companion flange of the universal joint. This flange is held on by a nut on the $\frac{3}{4}$ in diameter threaded end of the spindle. The nut is tightened until the pre-load on the roller bearings is such that the torque required to turn the spindle is 12 to 15 lb-in. The outer races of both bearings are pulled up against shoulders in the nose piece, and the inner ones are separated by a tubular distance

Adjustment of the axial position of the pinion is effected by means of washers of suitable thickness interposed between the gear and the inner race of the rear bearing. An oil seal is housed in the front end of the nose piece, and bears round the boss of the companion flange for the universal joint. This seal, which is supplied by the Super Oil Seals and Gaskets Ltd., is protected by a shroud ring pressed on to the boss of the companion flange.

An En 35A forged crown wheel with 40 teeth is employed. The outside diameter is 7.23 in and the inside diameter 406 in. It is drilled and tapped for eight $\frac{1}{16}$ in diameter set bolts which secure it to the inner face of the flange around the one-piece, black heart malleable cast iron differential cage.



The two differential pinions are of En 362. They are carried on a $\frac{8}{8}$ in diameter En 18B pinion spindle which is secured by a peg in a hole through the differential cage and one end of the spindle. Phosphor bronze, spherical thrust washers, $1\frac{1}{2}$ in outside diameter, are interposed between the outer ends of the pinions and the cage.

The En 35A forged differential gears are slightly more than $2\frac{1}{8}$ in diameter, and the length of the tooth engagement is approximately $\frac{1}{8}$ in. Flat, phosphor bronze thrust washers of $2\frac{1}{8}$ in outside diameter are employed, and the bearing length of the 1-32 in diameter gear boss in the differential cage is 0-56 in. The back lash between the differential gears and pinions is 0-005-0-007 in.

Two taper roller bearings carry the differential cage. They are spaced approximately 5 in apart between centres. Their inner races bear against shoulders on the cage, and their outer races are held by ring nuts screwed into the housings in the nose piece. These ring nuts control the mesh of the crown wheel and pinion, as well as the bearing pre-load which is gauged by measuring the spring of the bearing caps. The amount of spring should be 0-01-0-012 in.

The En 18A shafts are 1.06 in diameter at the ends, but they are reduced to 0.94 in diameter over most of the centre portion. Splines on the inner ends transmit the drive from the differential gears. The spline root diameter is 0.865-0.875 in, and the depth is 0.0548-0.0610 in. On the outer end of each shaft, a flange is upset to carry the cast iron brake drum, wheel and En 5B or En 8B forged bearing housing. A single row ball bearing is employed. Its outer face is clamped

between the housing and the upset

flange on the half shaft, and the inner

race is pulled against a shoulder on the hub by a ring nut tightened against its outer face. This nut is locked by a tab washer.

Two oil seals are employed, one in the bearing housing to prevent grease or oil passing from the bearing into the brake drum; the other is carried in the hub, and bears on the periphery of the half shaft to prevent the escape of oil from the axle. The hub and brake back plate carrier are an integral En 8B forging welded on to the axle tube which is 21 in outside diameter by 0-152-0.168 in. thick. The inner ends of the axle tubes are welded to the banjo Knock-on casing. wheel hubs may be fitted if required by the customer. They

are used in conjunction with the larger wheels which are wire braced. Although these hub assemblies are not Ford components, all parts such as bearings, seals, etc., subject to wear are common to both types of hub.



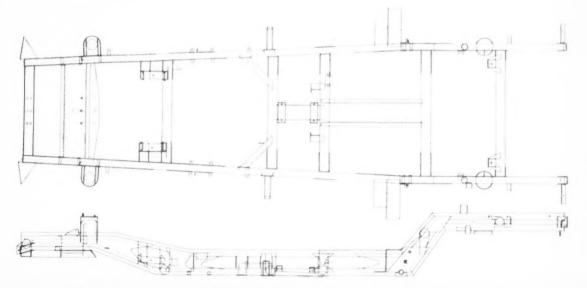
Coil springs are used on the rear suspension. With this arrangement it is necessary, of course, to provide positive longitudinal and lateral location. For this reason, almost parallel trailing links are employed in conjunction with a Panhard rod. The unsprung weight is 190 lb. At the wheel, the rate is 105 lb-in. It is obtained with a



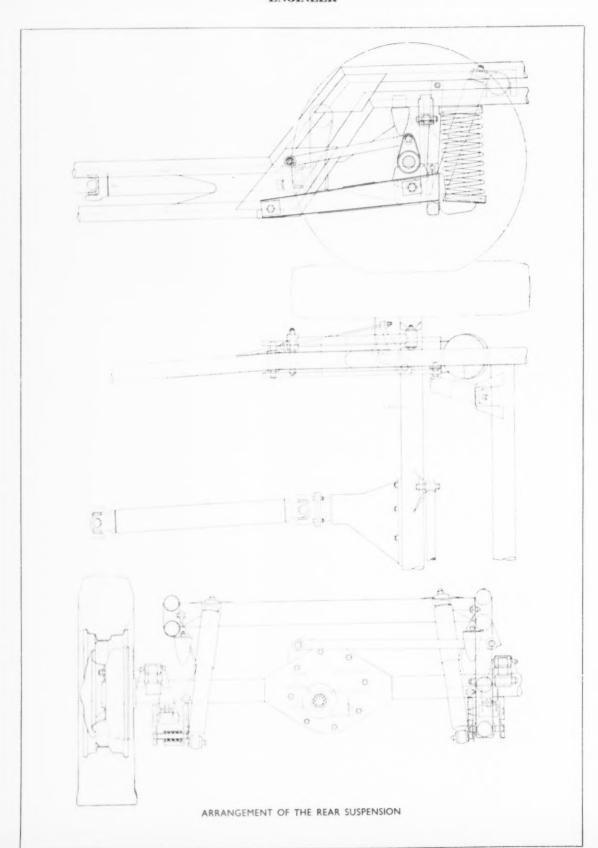
A Panhard rod, attached at one end to the frame and at the other to the differential casing, provides the necessary lateral location for the rear axis

spring that has a rate of 56 lb-in. This arrangement gives a periodicity of 92 cycles/min. The height of the roll, which is assumed to be the height of the lower bearing of the Panhard rod, is 16-5 in above the ground. The wheel deflection to fully laden position is 4-1 in, and to full bump 3 in.

On each side, the En 45A spring is mounted between two fabricated steel pans, one under the frame side member and the other on an extension of the lower trailing link. The free length of the spring is 16-19 in, and in the static fully laden position it is 10-5 in long. Its overall diameter is 4½—the wire is the diameter and there are 10-7



No pressings are employed in the chassis frame, the main members of which are constructed from steel tube



effective coils.

The length of the lower trailing link on each side is 151 in between pin centres, and that of the upper one 13k in. At the front end of each pair of links, the vertical spacing of the centres is 43 in; at the axle they are 54 in apart. Each lower trailing link is fabricated from 14 s.w.g. plates pivoted on the ends of a 1 in diameter, En 8 bolt. This bolt is carried in a 1 in diameter Silentbloc bush in a tube welded in the lower part of the frame side member. The upper and lower edges of each plate are flanged and turned outwards, and 14 s.w.g. spacer plates are welded to them. A 14 s.w.g. end plate is welded between the two side plates at the rear and two L-shaped brackets to carry the spring pan are welded on to it. The depth of the link is approximately 1 in at the pivot point and 13 in where it is attached to the axle. This attachment takes the form of another 1 in diameter En 8 bolt, also in a Silentbloc bush. bush is carried in a tube welded to a U-bracket bolted beneath a similar bracket which in the Ford cars supports the semi-elliptic springs under the axle tube. By using the existing brackets so far as possible, the need for welding on other fittings and the danger of distortion of the axle tube are avoided.

The upper trailing link is fabricated

from 1 in diameter by 8 s.w.g. tube with 14 in diameter by 8 s.w.g. tubular These end fittings are end fittings. welded on with their axes perpendicular to the major axis of the link. Silentbloc bushes, 14 in long, are fitted at both ends. The pivot end of each upper link is cantilever mounted on a in diameter En 8B bolt passed through the vertical side-plates welded on the inner and outer faces of the two tubes that form the frame side member. At the other end of the link, a similar bolt is passed through the bush which is carried between two 10 s.w.g. flanged vertical lugs welded on top of the axle tube. Careful checks have shown that the welding on of these two lugs does not cause any distortion.

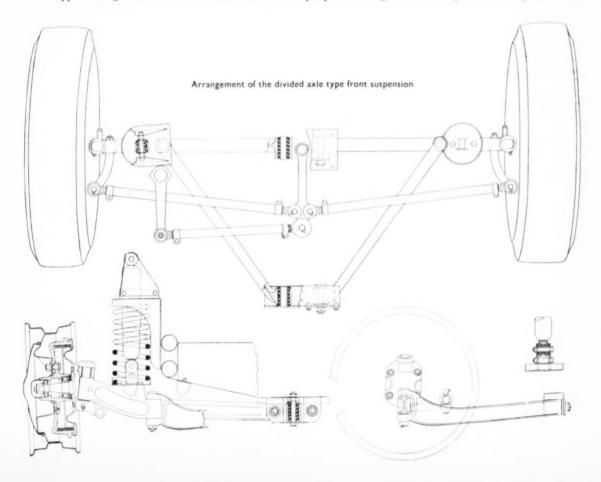
A 1 in diameter by 14 s.w.g. Panhard rod is fitted. Its centre-to-centre length is 211 in. Both ends are carried by in diameter En 8 bolts in Silentbloc rubber bushes in welded-on tubular end fittings which are 14 in outside diameter by 8 s.w.g. One end of the rod is overhung mounted on a bracket welded to the top of the banjo casing and the other is carried between two lugs on to the lower tube of the frame side member on the left-hand side.

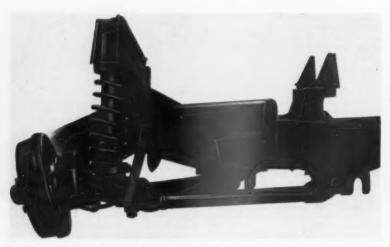
Armstrong AT7, 7 in bore telescopic dampers are employed. The ring-type lower end fitting is overhung-mounted on a 1% in diameter pin passed through

a vertical strip of 10 s.w.g. steel welded to the flanges of the inner plate of the lower transverse link. The pin is welded to both the vertical strip and the inner plate. One end of a Balata strap, which forms a rebound stop, is bolted to the top of this strip; the other end is secured to the frame side member. At the top of the damper, a rubber sandwich end fitting is employed. It is mounted on a 10 s.w.g. triangulated, U-section bracket welded to a frame cross member. Rubber rebound stops are bolted to brackets under the frame side members and they bear on top of the axle tube.

Front suspension and steering

A divided axle and coil spring front suspension layout has been adopted, and the drag loads are taken by forward extended radius rods. The axes of the pivot bearings of these rods are in line with those of the axle pivot bearings. The advantages of the divided axle layout are well known. It gives a high roll centre as compared with the double wishbone layout, and when the body and frame structure is rolling during a turn, the wheels take up a more favourable attitude relative to the road. It is claimed that this results in less tyre wear. From the static position to full bump the camber change is about 7 deg and the change in track is





In this illustration, the front end pivot of the radius rod is shown in a partly assembled condition

approximately $\frac{2}{8}$ in. The normal camber and castor angles are respectively $2\frac{1}{2}$ deg and 2 deg while the swivel pin angle is 7 deg. A toe in of $0-\frac{1}{8}$ in is incorporated.

The wheel deflection to the static laden position is 8.25 in, and to full bump 11.25 in. With the vehicle in the normal laden state, the height of the roll centre above the ground is 10.75 in. The rate at the wheel is 52 lb/in and at the spring 140 lb/in. This gives a periodicity of 65 cycles/min with the Zephyr engine installed, and 68 cycles/min with the Consul. The unsprung weight is 66] lb on each

A co-axial coil spring and telescopic damper arrangement has been adopted. The spring is made from $\frac{1}{16}$ in diameter, En 45A ground bar. It has a free length of 13-4 in, and its installed length in the fully laden position is 8-5 in with the Consul, and 8-0 in with the Zephyr engined chassis. The overall diameter of the spring is 4 in and there are 8-34 effective coils. At the upper end, the spring bears on a fabricated steel pan welded to the frame side member. The Armstrong AT7 damper passes through a clearance hole in the centre of the pan. The

ring type end fitting at the top of the damper is rubber bushed and carried about 41 in above the spring pan by a kin diameter bolt between two 14 s.w.g. lugs. Each of these lugs is bolted to a 14 s.w.g. right angle bracket welded on top of the pan. arrangement is necessary in order that the appropriate axial compression may be applied to the bush. The lower end of the spring rests on a 14 s.w.g. pan bolted to a boss on top of the divided axle. Another rubber bushed ring type end fitting at the bottom of the shock absorber is carried in a similar manner to the upper one between two 10 s.w.g. lugs bolted to a 10 s.w.g. U-bracket welded to the lower spring pan. The effective radius of the line of action of the spring and shock absorber is 151 in from the pivot point of the swing axle.

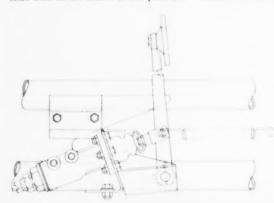
An En 18A I-section forging forms

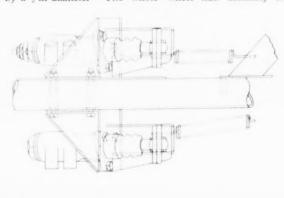
An En 18A I-section forging forms the divided axle. Its effective radius measured to the wheel centre is 25 in. The inner bearing is a Silentbloc bush, and is carried on a \(\) in diameter mild steel pin 2 in from the centre line of the chassis, in a 10 s.w.g. inverted U-bracket. About 15 in outboard of the pivot point, the forged fork-end fitting of the tubular radius rod is secured to the axle by a \(\) in diameter

pin. In a tubular fitting at the other end of this rod, there is a Silentbloc flanged bush with its axis in line with, and its centre about 16 in in front of, that of the swing axle pivot bush. A in diameter mild steel bolt carries the radius rod bush in a 14 s.w.g. inverted U-bracket bolted to the underside of the frame cross member. An inverted U-shaped cut-out in the front face of this bracket clears the bush when it is assembled and a separate bolted on plate is fitted in front of it to apply the compressive pre-load. The rubber bump stop is mounted just inboard of the side member under the cross member supporting the axle pivot bearings. At full bump it bears directly on the axle. The rebound stop is formed by a Balata strap passed under the axle just outboard of the point of attachment of the drag link. The ends of the strap are bolted to the lower ends of the side plates which form gussets between the upper spring pan and the outer face of the frame side member.

A 1 % in diameter, En 207 swivel pin is carried in a boss on the outer end of the axle, and is secured by a cotter pin. This pin is passed through the axle boss and engages in a groove in the swivel pin. Phosphor bronze bushes, spaced 21 in apart in bosses on the stub axle forging, form the upper and lower swivel pin bearings. These bushes are 1 in long. They are lubricated through a grease nipple in the lower bearing boss and drillings in the pin. The thrust is taken by a ball thrust bearing between a head on the swivel pin and the top face of the upper bearing boss. A felt seal is interposed between the upper face of the bottom bearing and the axle.

The stub axle forging is of En 18A. Two taper roller bearings carry the wheel, the inner bearing being mounted on the 1½ in diameter portion of the stub axle and the outer one on the ½ in diameter portion. The two bearings are spaced approximately 1½ in apart. They are assembled from each end into the wheel hub. A Superfelt "G" type grease seal is carried in the inner end of the hub and bears on a 1½ in diameter portion of the stub axle. The whole wheel hub assembly is





The brake and clutch master cylinders are mounted on brackets on each side of a frame side member

pulled on to the stub axle by a \(\frac{5}{8} \) in nut and washer. The inner race of the rear bearing is against a shoulder on the stub axle and the two outer races bear against shoulders on the hub. A pressed steel cap over the outer end of the hub serves to retain the grease. If any grease should escape past the seal on the inner end, it is caught in a trap formed by a shroud ring surrounding the inner end of the hub and bolted to the brake back plate. The brake drum is secured in the conventional manner to a flange around the hub.

Marles Hourglass worm and roller steering gear, manufactured by Adamant Engineering Co., Ltd., of Luton, is fitted. The steering box is mounted on the side frame approximately 6 in in front of the swing axle. The ratio is 14:1, giving 2\frac{1}{4} turns from lock to lock. On front lock, the wheel angle is 34 deg, and it is 36 deg on the other lock. This gives a turning circle of 28 ft. A 17 in diameter steering wheel is employed.

A divided track rod system has been adopted. Thompson or Ford adjustable ball joints are fitted throughout. They are lubricated by grease nipples. The ball joints are screwed into the split ends of the a in diameter by 9 s.w.g. tubes. The setting is fixed by tightening a bolt passed through a clamping ring round the tube. At their outer ends the track rods are each connected to an En 18A steering arm, which is a push fit in a lug on the lower swivel pin bearing boss. The arm is retained in the lug by a split pinned slotted nut on its end. effective length of the arm is 5 in.

The track rods are $21\frac{1}{6}$ in long. At their inner ends they are attached to an idler lever, at a point $6\frac{1}{2}$ in from its pivot. The pivot pin in this lever is $\frac{1}{6}$ in diameter and is carried in a Silentbloc bush, $1\frac{1}{6}$ in long. At the outer end of the idler lever is attached the connecting link to the steering drop arm. The effective radius from the line of action of this link about the

idler pivot centre is 81 in, and that of the drop arm is 61 in.

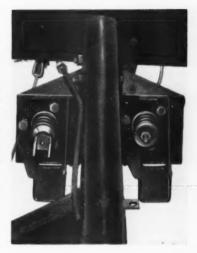
Brakes

Girling two leading shoe hydraulic brakes are fitted at the front, and two trailing shoe units are employed at the The cast iron brake drums are 9 in diameter, the shoe width is 11 in and the friction lining area is 121 sq in. A pistol grip type hand brake is mounted under the dash. It operates a lever pivoted on a bolt in a bracket on a frame cross member under the engine. A lever ratio of 12:1 has been adopted. The brake control cable is pinned to the lower end of the lever and passed back and attached to the centre of a cross link compensator. Each end of the cross link is pinned to a Bowden cable control. These controls are connected to the bell crank brake operating levers mounted one on each end of the axle tube.

A mild steel pivot pin with a phosphor bronze bush, $\frac{1}{8}$ in inside diameter by $1\frac{1}{2}$ in long, carries the lower end of the brake padal. This pin is in a bracket on the inner face of the frame side member. The master cylinder is bolted to the same bracket, and the fork end of its plunger is attached to the pedal stem at a point about $2\frac{1}{16}$ in above the pivot to give a lever ratio of 4.25:1. A similar bracket on the other face of the frame side member carries the clutch pedal and master cylinder. The principal dimensions and the lever ratio of the clutch control are approximately the same as those of the brake unit.

Frame

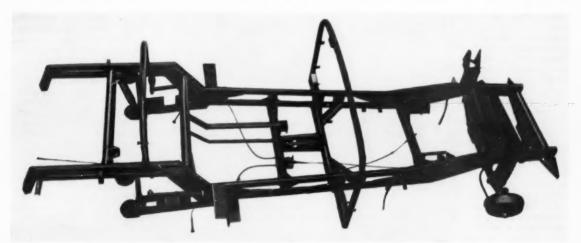
A notable feature of the frame is that in order to avoid the use of pressed steel components it is built up almost entirely of tubular members. As a result, it is possible to manufacture and assemble all parts of the frame at the Clapham factory. Moreover, should modifications be required in the future, to adapt the frame to different vehicles, there are no undue restric-



Brake and clutch master cylinders

tions imposed by the necessity of recovering the high cost of press tools. In order to localize distortion, almost all the welded joints are arranged parallel to the axis of the tube.

Seven cross members are incor-orated. All except the main cross porated. member that supports the suspension are 21 in diameter by 16 s.w.g. mild steel tube. The first carries the bearings for the radius rods of the front suspension. The second is the cross member supporting the axle and suspension components. It is fabricated as follows: Two plates vertically positioned and flanged outwards at their top and bottom edges form the front and rear walls which are about 5½ in apart. A flat plate is welded up to the bottom two flanges, and another is welded horizontally about 14 in above it between the two vertical plates. Thus, a box section is formed beneath the second horizontal plate and a channel section above it. The swing axles are pivoted in an inverted U-bracket at the centre of the member



The frame is of sturdy construction

under the lower plate beneath the rear wall.

The next cross member to the rear helps to support the brackets for the front engine mountings. These brackets are welded to the cross member and the frame side members. Then there are two cross members underneath the seats. Additional support for the foremost of these two members is afforded by box section gussets to the side frames. Welded beneath these two members at the centre line of the chassis is a longitudinal inverted channel section. It carries two brackets for the engine mounting points under the gearbox rear extension. Two more brackets, one on each side of the longitudinal channel, are welded on the rearmost of the two cross members to carry the gear shift mechanism. Two longitudinal pieces of angle section extend back from the rearmost cross member to a bracket suspended under another cross member where the frame is cranked upwards to clear the rear These two angle pieces support the propeller shaft tunnel. Another cross member is positioned approximately 4 in behind the axle and two 10 s.w.g. brackets are welded to it to carry the telescopic shock absorbers.

The frame side members are each fabricated from two 13 in diameter by

16 s.w.g. mild steel tubes positioned one above the other and spaced apart by 14 s.w.g. or alternatively 16 s.w.g. These plates are positioned on each side member as follows. Two are welded, one on each side, at the front, where the members are bent to form an upward crank so that the front end is approximately 4 in above the centre. The outer plate extends back to the cross member supporting the front engine mountings, and the inner one to a point about 151 in further to the rear. Immediately to the rear of this cross member a similar plate, but of I-shape, is welded on the outer face of each side member.

Five more interconnecting plates are welded between the tubes in the centre portion of the frame where the axes of the tubes are 5 in apart. Three of them are on the inner face, two being U-sections into which are welded the ends of the two tubular cross members, while the third is flanged outwards, and supports the gusset which is welded to its flat inner face to support a cross member. One of the two plates on the outer face of each side member is flanged inwards and carries a body mounting outrigger bracket, while the other is an I-section plate posi-tioned in line with the rearmost of the two centre cross members.

Near the back, the frame is cut, mitred and welded together again to form an upward crank to clear the rear axle. A 14 or 16 s.w.g. plate is welded on the inner face of each side member from the rear cross member forwards to about 12 in in front of the lower part of the crank. Two more plates are welded on, one at the lower crank and the other at the upper one. The rear ends of each pair of tubes are welded inside short vertical channel sections. Carried on the outer face of each channel is a body mounting bracket.

The 14 s.w.g. spring pans are welded under the side members at the rear and gusseted to the inner plate. At the front they are welded and gusseted to the outer plate. Three 14 s.w.g. Z-section outrigger brackets for the body mountings are carried on each side frame. These are the two already mentioned, one in line with the foremost of the two centre cross members, the other at the extreme back, and the third is at the lower portion of the rear crank. Two more brackets, one on each side at the extreme front end, carry the bumper irons. Two hoops, $1\frac{1}{2}$ in diameter by 16 s.w.g., are mounted on the frame, one under the scuttle and the other under the rear decking.

CARBIDE PLATING

TUNGSTEN carbide is now estab-lished as a hard and wear-resistant cutting tool material; but of recent months a process has been introduced which offers further potentialities. This is the plating of metals with tungsten carbide. The deposit is in the order of 0.005 in. to 0.020 in., and has been applied to steels of every kind, cast iron, aluminium, copper, brass, bronze. titanium and magnesium. There are, however, a number of metallic surfaces that will not, under present conditions, accept this plating.

The principal feature of the process is that the temperature of the basis metal remains below 200 deg C while plating is being carried out, so that it is not subjected to high heat stresses, with consequent distortion, nor is the work injured in any way. Up to now, the maximum area plated is 6 in wide by 40 in long, and the plating can be applied to cylindrical surfaces, flat surfaces, the interiors of holes, and formed or contoured parts.

The object of applying the plating is to confer the wear-resistance of solid sintered carbide, which is much greater than that of chromium plate, steels, etc. In addition, the plated parts have a lower elastic modulus and a higher degree of resistance to impact and thermal shock.

Another use for carbide plating is in the protection of precision parts, when the abrasion or frictional wear encountered has to reduce the dimensions by only a few thousandths of an inch to make the part fit only for the scrap heap. The coating can be worn down to the base metal and then renewed.

In powder metallurgy, use is made core rods of sintered tungsten carbide. These have the disadvantage that they are low in ductility. If plated carbide rods are substituted for these, there is an immediate gain in increased resistance and toughness. It should be noted also that the wide range of basis metals capable of taking the plating affords interesting opportunities to the designer and engineer. For example, copper can be plated so that in addition to high electrical conductivity, it has greatly increased resistance to wear. Again, the carbide plating of aluminium alloys enables weight to be saved without difficulty in applications where high degrees of frictional and abrasive wear would render the use of these alloys undesirable in normal circumstances.

The carbide employed in the new process is not identical with the sintered carbide used as tips in cutting tools and other parts. The material is composed of 8 per cent cobalt together with the carbides of tungsten, in various forms. Cobalt is not free. The coating process produces a coating made up of particles of longitudinal form and disc-like shape, with diameters far exceeding thickness. The coating forms a purely mechanical interlocking bond with the basis metal. It does not form a welded bond, since the particles of the carbide do not interpenetrate or alloy with the basis

There is no great difficulty in preparing work for carbide plating. parts must be undercut to a depth sufficient to suit the plating thickness. A small protective lip should be provided at any edge subject to excessive wear or careless use or wherever two carbide-coated surfaces form an angle of 90 deg with each other. Usually the plating does not exceed 0.001 in thick over the permissible wear limit, as this gives maximum resistance to mechanical and thermal shock; much more so than a coating of, say, 0.01-0.02 in. When the work needs to be finish-ground, an excess plating of 0.004 in is put down. Finish-grinding develops a surface of 2 micro-in r.m.s. For this grinding operation it is advisable to employ a resinoid-bonded diamond wheel. For rough grinding, a 100-mesh wheel is required, and for such operations as call for the removal of only 0.0005 in, a 400-mesh wheel.

An interesting point is that a comprehensive test of wear performed on a wide range of different materials clearly demonstrated that a carbideplated gauge had superior wearresistance to all the others. medium providing wear was cylinderblock cast iron. The gauge proved to last approximately five times as long as boron carbide and three times as long as solid sintered tungsten carbide made up into gauges. The expense of coating with tungsten carbide depends mainly on the dimensions and form of the work, the output required, and the plating thickness.

COLD EXTRUSION

A Résumé of American Developments

SOME small steel parts for motor vehicle mechanisms are being produced now from cold extrusion pressings. This and other applications of the process is an important new development, for steel is one of the least plastic materials when cold and almost universally is formed by forging, pressing or extruding while heated to high temperature to reduce resistance to deformation. Cold extrusion is still in the experimental and development stage, however, with press manufac-

turers and industrial production companies devoting much study to the numerous problems involved. But for certain applications it has been used successfully on a large scale for several years and research promises to make it more widely applicable and economically advantageous.

The evolution of the process dates back to World War II, when the Germans built some special presses for the production of artillery shells. They adopted this process save labour and avoid waste of highgrade processed steel in the form of machining scrap. One of the presses was brought to the United States after the war and the Office of Technical Services of the Department of Commerce reproduced a detailed description of the extrusion technique written by the German engineers. This was issued in 1946 under the title PB 39371 "Cold Shaping of Steel." It consists of 172 pages with drawings and is available from the U.S. Library of Congress, Photo - duplication Service Board, at a unit price of \$4 for a microfilm reproduc-tion or \$12 for a photostat copy.

United

Several United States press manufacturers have designed and built special mechanical and hydraulic presses which are being used in the production of ordnance shells in sizes up to 155 millimetres weighing 76 pounds. One of the largest mechanical presses, built by the E. W. Bliss Company, weighs 305 tons (671,000 pounds) and can exert a pressure of 4,000 tons (8,800,000 pounds). Presses of smaller size and less capacity are used for extruding smaller shells, explosive rockets and automotive parts such as ball bearing races, special bolts, hollow shafts, pistons and motor cycle engine-starter

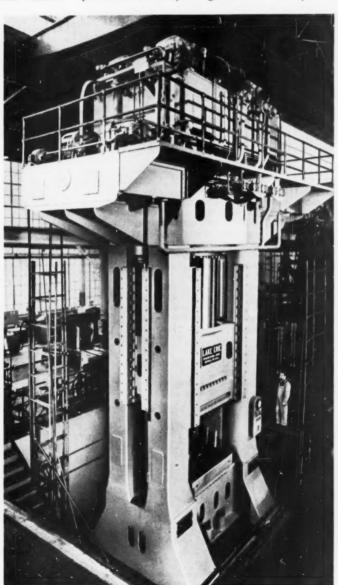
gears. All these presses must be strong and rugged enough to withstand without damage to their structure the reaction forces from the extreme pressure that cold steel extrusion requires. The principal advantages of the pro-

The principal advantages of the process are that billets or slugs of cheaper grades of steel, such as low-carbon (0-10 per cent or less) or spheroidized higher-carbon types, can be used in place of heat-treated material; preheating is eliminated; extruding refines the crystalline structure and improves

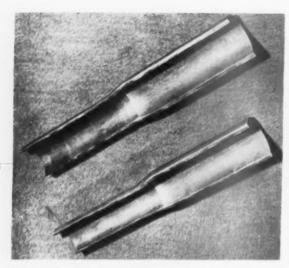
the hardness and toughness; extrusions need little or no machining; creation of turning ribbons and boring chips can be virtually eliminated; rate of output is greatly increased, and cost of labour substantially reduced.

These advantages can be realized, however, only if the quantities of items produced are enough to offset the high cost of the presses compared with customary forging or drawing equipment and the special dies required for extruding operations. The Bliss company states that the stresses built up in making cold steel flow in steel dies are so high (they may be up to 150,000 lb/in2 or more than double that figure) that it is desirable to draw or stress relieve the tool members at fairly frequent intervals to counteract fatigue of the tool metal. It recommends also the use of carbide inserts in dies at areas of greatest pressure. Such inserts should be supported on hardened steel blocks and held in place by tapered shrunken holders.

Dies are of two general types: open dies, in which the metal does not fill the cavities during extrusion; and closed dies, in which the metal fills all corners and other recesses. The latter type is said to



Lake Erie 3,000 tons hydraulic press for cold steel extrusion



in diameter steel tubing showing the reduction that can be effected in a single pass

require extreme pressure to overcome resistance unless relief to metal flow is provided at one or two places.

Pressure needed depends upon the yield point of the metal before extrusion, frictional resistance imposed by dies to flow of the metal, speed of extrusion, and desired yield point of products after extrusion. If flow resistance and pressing speed are low, the amount of pressure can be based on yield point after work hardening. Restricted flow and high speed may triple the power load on the pressure required on grade C-1010 steel ranges from 100,000 to 330,000 lb/in², and on spheroidized C-1020 steel from 120,000 to 400,000 lb/in², depending upon various factors. Pressure of 314,000 lb/in² was necessary for a "backward extrusion" operation on a steel billet 4 in in diameter when using an open die and a piercing punch with a cross section area of 5-9 in.

with a cross section area of 5.9 in.

The term "backward" or "upward" extrusion, as described by the Federal

Machine & Welder Company, refers to an operation in which a punch descending into a die closed at the base causes the work metal to flow upward along the surface of the punch as the punch forces the metal outward from the centre of the slug and reduces the cross - section area. In the other type of extrusion, termed "forward" "downward," a cup formed either by the "backward" method or by hot forging is placed in a die and a punch smallhaving a diameter pilot end

enters the open end of the cup. As the punch descends, the pilot enters an orifice in the bottom of the die and the larger part of the tapered punch presses against the inner wall of the shell, reducing the thickness and extending the length downward.

When pressure stress within the metal becomes as high as the tensile yield point, slippage within the metal is much harder to force and greatly increased pressure is necessary to effect further reduction in the wall thickness or cross-section area of an extrusion. Alloyed steel is more resistant to pressing than pure or plain low-carbon steel because alloys bind the ferrous grains and impede internal slippage along adjacent planes. The hardness of metals before extrusion should not exceed Rockwell B 60 if best results are to be obtained, according to the Bliss company. Harder steels can be used but only at considerable sacrifice of the amount of area reduction.

Friction between the metal, punch

and die surfaces is high in most work. Ordinary lubricants serve to overcome most of it if conditions are favourable but may break down, allowing frictional heat to cause cracking of the surface of extrusions produced. The most effective lubrication method developed is to bond a layer of phosphate to the steel and apply a coat of phosphate-bonded high-pressure stearate lubricant. The phosphate serves as a parting layer that minimizes contact of metal with metal and the tendency of the surfaces to weld together or pick up abraded particles from each other.

But the Mullins Manufacturing Corporation, which has been producing large quantities of ordnance shells by cold extrusion for several years, and various automotive parts more recently with fast standard Lake Erie hydraulic presses of 2,000 and 3,000-ton capacity, uses neither phosphate coating nor special lubricant. It has found that ordinary bonderizing treatment and usual deep-drawing lubricating compounds suffice if die and punch designs are right. Cross-section area reductions up to 98 per cent have been accomplished without using any special lubricants.

Extrusions can be made from circular, hexagonal or round-cornered square slugs, but dies should not have sharp corners. If die and punch design and the setup in the press are correct, the rate of extrusion or the per cent of area reduction is said to be limited only by work hardening of the metal. Mullins has found that annealing or normalizing becomes necessary when the hardness of any type of steel reaches Rockwell B 100. After the hardness is reduced the steel can be worked again at maximum speed until it hardens once more. Extruding has been done at a rate of 5 in/sec in the 3,000 ton standard hydraulic press.

The pressure that may be applied is not limited by the strength of dies, since these can be made with ample metal and holder support to resist the forces, but diameter and length of the



Ball bearing races extruded from a single piece of chromium steel



Starter gear for motor-cycle engine, 2 in O.D. × 1½ in long

punches are limiting factors in the extruding of tubular objects such as artillery shells. Too high pressure will generate heat that may damage the surface of the punch or even cause it to break. Punches made by Mullins from tool steel heat-treated to Rockwell C 65 hardness withstand pressure up to 275,000 lb/in2. Several successive pressings with different punches are necessary to deform a solid slug of steel into a thin-walled ordnance shell. Movement of the punches must be at a uniform rate and without stoppage or the metal will harden rapidly. excessive pressure needed to restart the operation may break a punch or a die

For production of 4½ in diameter nose-pieces for rocket projectiles, the Pontiac Division of the General Motors Corporation uses slugs 5½ in in in diameter and 2½ in long annealed in a salt bath at 1,250 deg Fahrenheit and cleaned of scale by shot-blasting. A coat of phosphate is applied to the slug, which is then immersed in liquid soap of 140 deg temperature before extrusion in a Clearing mechanical press of 2,500 tons capacity. Backward extrusion

reduces the transverse area about 68 per cent in forming a rocket nose cup about 6 in long, $5\frac{1}{2}$ in in diameter and with a wall $\frac{1}{2}$ in thick. The cooled cup has a Rockwell B hardness of approximately 100 to 105. Pressing may raise the temperature of the cup to 400 or 500 deg unless the die and punch are designed to reduce surface friction to a minimum.

Design of the die, die holder, punch and operating construction is somewhat complex. It includes a punch stripper plate with pneumatic means for return of the plate after withdrawal of the punch; a die ring 12 in in diameter and 5 in wall thickness pressed into a cast alloy-steel holder plate 12 in thick; an ejector plug in the die pressure shoe, and an ejector pin for raising the plug and forcing extruded nose cups out of the die. Punches and dies are made of high-speed alloy steel heat-treated for maximum resistance to scarifying.

Pressing of the cups as described is the first of a sequence of operations performed in drawing out cold steel slugs to the required length and wall thickness of the rocket nose-piece destined to contain an explosive. It indicates how the process may be adapted to production of some simple parts for motor vehicles and various stationary machines.

Cold extrusion not only avoids

Cold extrusion not only avoids expenditure of labour in lathe turning and boring forged billets, and consequent waste of expensive metal in the form of scrap, but forms products to close tolerances and with so smooth a surface that no machining or other finishing operations are necessary if the die surface is smooth enough. The process also imparts ductility and impact strength equal to that of tempered alloy steels up to 110,000 lb/in² and avoids danger of cracking and distortion by heat treating.

Development of the process is advancing fast, with press manufacturers doing research and experimenting, and industrial users devising individual techniques for producing their different products of high quality with efficiency. This accounts for differences in practice, but all are looking forward optimistically to future large-scale application of the process to production of parts for construction of machines for civilian use.

RESEARCH EQUIPMENT

New Apparatus Developed for Engine Research

THREE new pieces of apparatus for engine research are described in the March 1953 issue of Engineers' Digest. The first is a quick-acting gas sampler. It has been developed to make possible the sampling, at a given instant, of gas in petrol and oil engine combustion chambers. This sampler was designed for use in research on the mechanism of engine knock, and with the object of obtaining information relative to the more efficient use of automotive fuels.

To obtain significant data on the chemical processes taking place in the cylinder of the engine, some knowledge is required of the proportion of reactants and products present at various times during the combustion Multi-stage combustion reactions are especially difficult to study because of the very short duration of each stage, varying as it does between 0.1 and 1.0 milliseconds, the extreme complexity of the reaction mechanism, and the fact that many of the important constituents are free radicals with half-lives of one millisecond or less. Thus, the samples must be taken over extremely short intervals during a single cycle of operation, and the reaction must be frozen as completely as possible upon removal of the sample from the combustion chamber.

The principal parts of the sampling device are a flanged piston valve and an evacuated chamber, in which the valve operates. At the upper end of the valve chamber is a connection to an evacuated sample container and a vacuum pump. The lower end of the sampler is threaded to fit a sparking

plug hole. An electrically-operated trigger mechanism releases the valve and gases pass into the sample container until the valve has reached its extreme position and has sealed the container.

A three-dimensional flow measuring probe is the second piece of apparatus described. It has been designed to satisfy the need for a device to facilitate the study of fluid flow, especially in combustion chambers, compressors and diffusers. The head of the probe is a sphere which is attached to a support tube in such a way that it has two degrees of freedom, one of translation along the axis of the tube and the other of rotation about it. In this respect, the instrument is similar to the two-dimensional Pitot static tube with which it is impossible to measure flow that does not lie in a plane at right angles to the support tube axis. new probe, however, is a three-dimensional type instrument in that the pitch and yaw angles of the velocity vector are detected and measured by holes on the surface of the sphere. Yaw angle can be determined through the full range of ±180 deg and the pitch angle through a limited angle of ±45 deg. The instrument could be modified to give readings for greater pitch angles. Static pressure is observed directly from readings of manometers or other pressure measuring devices connected to the instrument, and dynamic pres-sure is determined from the observed pressures at the several holes on the surface of the sphere.

In numerous heat transfer investigations, the need arises for an instrument to measure transient temperatures. Because of their simplicity and comparatively rapid response, fine wire thermocouples are usually employed. However, in certain applications, this type of instrument is unsatisfactory because of its lack of strength and the difficulty of positioning the junctions at the point to be investigated. Furthermore, the minimum size of the junction, which affects the rate of response, usually is limited to the wire diameter.

A special thermocouple has now been developed which has a junction thickness of only one micron, that is, one-thousandth of a millimetre. Yet the instrument is said to be of such rugged construction that it may be employed for measuring extremely rapid temperature changes in regions of high stress.

The major parts of the instrument are the probe, body, terminal posts and an insulator. The probe consists of a thick walled steel tube, one end of which is plated with a layer of nickel, one micron thick. Between the nickel plating and the end of the steel tubing is the plane at which the thermal electromotive force is generated. A lead wire to the plating is carried in the tubing.

A steel body, which is externally threaded for mounting, carries the probe. This body also contains the terminal post. The overall length of the couple is approximately 3 in. In a later design, the probe itself is threaded and the nickel wire is connected directly to the external lead so that neither the insulator nor the body is required. (2043)

FUEL CONSUMPTION

An Investigation into the Effect of Benzole

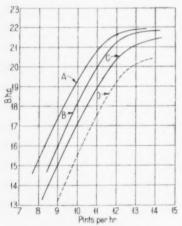
by L. W. Barnard, B.Sc.(Eng.), A.C.G.I., D.I.C. and G. Claxton, M.Sc.

CONVENIENT starting point for discussions on this subject is an important fact demonstrated by Ricardo in 1923. An extensive series of tests then carried out showed that, provided the engine was being run on fuels of sufficiently high octane number to prevent knocking and that the mixture strength and ignition were adjusted for maximum performance, the power output and thermal efficiency were substantially the same for all except alcohol fuels. This means that the volumetric consumption of fuel is inversely proportional to the net calorific value per gallon. This value should be corrected by adding the latent heat of the fuel, because, when there is no application of external

heat to the carburettor, the volumetric efficiency will increase according to the latent heat of the fuel. On this basis, the combined effects of calorific value and latent heat should produce a saving of about 9 per cent for benzole in comparison with petrol on a volumetric

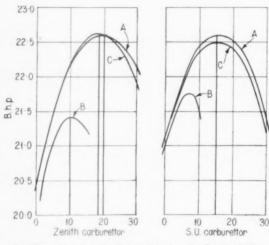
This statement needs many qualifications before it can be applied, even approximately, to fuel consumptions obtained during everyday conditions of service. The chief qualifications are:—

(1) Knock rating. Before the introduction of the present premium grade



A — Benzole mixture, ignition 20 deg B.T.D.C. B — 80 octane petrol, ignition 20 deg B.T.D.C. C — 70 octane petrol, ignition 10 deg B.T.D.C. D — 70 octane petrol, ignition T.D.C.

Fig. 2. Fuel consumption curves, Zenith



A — 80 octane petrol B — 70 octane petrol C — Benzole mixture Fig. 1. Ignition advance curves. 2,000 r.p.m. full throttle

fuels, most modern engines knocked violently on the fuel available, except when the ignition was considerably retarded. The change to premium grade fuels has allowed considerable ignition advance, so that more power is produced and there is a consequent saving in fuel.

(2) Mixture strength. power is generally obtained when the mixture strength supplied to the engine is about 20 per cent rich. Most carburettors are adjusted to supply such a mixture when the car is operating under steady conditions on petrol. Benzole, however, requires approximately 8 per cent more air per gallon for complete combustion than does petrol. It might, therefore, be thought that benzole or benzole mixture would be excessively rich unless some carburettor adjustment were made. However, much of this compensation is made automatically, because the effect of the higher specific gravity of the benzole is to cause a lowering of the level in the main jet. The effect on power output, and thus on fuel con-sumption, will also depend upon the shape of the power consumption curves (which coincide only at the point of maximum power), upon the mixture strength supplied, and upon the proportions of the mixture supplied by the main and compensating jets respectively.

Actual road tests to determine fuel consumption with different fuels give useful information, but the results are masked considerably by the following disturbing factors:—

(1) On successive runs the power output of the engine may vary owing

to differences in temperature, barometric pressure and humidity.

(2) Even when all the tests are made over the same route, the load may vary according to the weather conditions, particularly in so far as they affect wind resistance and road surfaces.

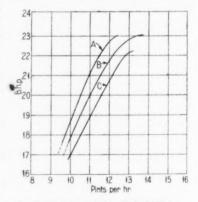
(3) Traffic conditions may cause variations in the necessity for gear changing, with consequent variations in both power requirements and throttle opening.

It is not surprising, therefore, that even when special arrangements are made for measuring accurately small quantities of fuel so that several runs can be made over a short route in one day, the variations between

repeat runs are often as great as the variations between It has also been different fuels. observed that whereas the change from one fuel to another may show a considerable difference in consumption for one make of car, the difference may be negligible for another make. Whilst the fuel consumption must depend to some extent upon distribution and the regularity of ignition setting for each cylinder, it is suggested that the differences are mainly due to the carburettor setting and the type of carburettor.

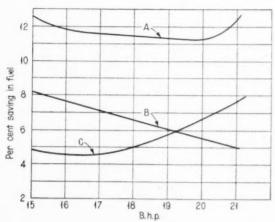
Experimental investigations

Because of the difficulties mentioned above, it was thought that more comparable and instructive data for the effect of fuel composition on consump-



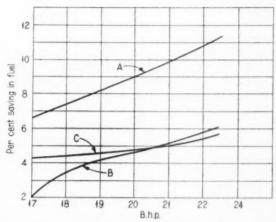
A — Benzole mixture, ignition 15 deg B.T.D.C. B — 80 octane petrol, ignition 15 deg B.T.D.C. C — 70 octane petrol, ignition 8 deg B.T.D.C.

Fig. 3. Fuel consumption curves, S.U. carburettor, optimum ignition



A — Benzole mixture/70 octane petrol B — 80 octane petrol/70 octane petrol
C — Benzole mixture/80 octane petrol

Fig. 4. Comparative fuel saving, Zenith carburettor



A — Benzole mixture/70 octane petrol. B — 80 octane petrol/70 octane petrol
C — Benzole mixture/80 octane petrol

Fig. 5. Comparative fuel saving, S.U. carburettor

tion would be obtainable from bench tests, since these would allow all the conditions to be controlled. An Austin A.40 engine was used in conjunction with an electric dynamometer. Two series of tests were carried out, each with three fuels. For one series the standard Zenith carburettor supplied with the engine was used, and for the other an S.U. carburettor that is recommended as a suitable alternative for

this engine. For all the tests the centrifugal automatic advance, but not the suctionoperated advance, was in operation. Preliminary tests were carried out with each fuel so that the static ignition setting could be adjusted to give the maximum power output at 2000 r.p.m. with full throttle. Fig. 1 shows that the optimum setting of the ignition is a function of both the fuel and the carburettor, although the same setting is suitable for either of the high octane fuels. The curves also show that at the maker's setting for the A.40 engine, that is, T.D.C., there is little extra power to be gained by using the higher octane fuel. Proper adjustment of the ignition for the higher octane fuel increases the power output above that obtained at the maker's setting by over Zenith 10 per cent with the carburettor.

All subsequent tests were run at 2000 r.p.m. with the appropriate ignition setting for each fuel; the power output was varied by varying the throttle opening. An engine speed of 2000 r.p.m. approximately corresponds with a road speed of 30 m.p.h., and the power outputs were those that would be necessary to drive a car at 30 m.p.h. up gradients between 1 in 20 and 1 in 10. It is recognized that for a large proportion of the running time an output of about 10 h.p. is sufficient to propel the vehicle at 30 m.p.h. Nevertheless, the tests are comparative and realistic, whereas the conventional tests, in which power consumption loops are obtained by varying the mixture strength supplied to an engine operating at full throttle and constant

speed, have no significance in terms of road use.

On the completion of the tests, both brake horse-power and fuel consumption in pints per hour were plotted against throttle opening. From these curves the consumptions for equal power outputs, irrespective of throttle opening, were determined and Figs. 2 and 3 were built up from the determinations. They show the consumption as a function of power for each of the fuels studied. The relative change in consumption at a particular power output is strictly proportional to the relative change in consumption in terms of miles per gallon. For comparative purposes, the power-con-sumption curve for 70 octane petrol used with the Zenith carburettor and with the maker's ignition setting of T.D.C. has been included in Fig. 2.

The fuels used were:-

(a) 70 octane petrol. This was a standard Pool grade with the following characteristics:

Sp. Gr. at 15.5 deg C. 0.720 Aromatic content, I.P.3/42

8.9 per cent volume 10.8 per cent weight

Octane No, M.M. 69-6 Octane No, R.M. 71-6

(b) Premium grade petrol. This was a leaded, premium grade petrol containing no added benzole and with the following characteristics:

Sp. Gr. at 15-5 deg C. . . 0-713 Aromatic content, I.P.3/42

15-3 per cent volume 18-7 per cent weight

Octane No, M.M. . . . 80-7 Octane No, R.M. . . . 89-6

(c) Benzole mixture (33 per cent benzole) with the following characteristics:

Sp. Gr. at 15.5 deg C. . . 0.787 Aromatic content, I.P.3/42

44-2 per cent volume 47-0 per cent weight

Octane No, M.M. . . . 80-4 Octane No, R.M. . . . 89-4 The test conditions were:-

1st series

Carburettor .. Zenith standard for A.40 engine

Engine speed . 2000 r.p.m. Throttle opening Varied

Ignition (static)

70 octane petrol . 10 deg B.T.D.C. Premium petrol . 20 deg B.T.D.C. Benzole mixture . 20 deg B.T.D.C.

2nd series

Carburettor .. S.U. replacement for

A.40 engine Engine speed . 2000 r.p.m.

Throttle opening Varied Ignition (static)

70 octane petrol . . 8 deg B.T.D.C. Premium petrol . . 15 deg B.T.D.C. Benzole mixture . . 15 deg B.T.D.C.

From the results shown in Figs. 2 and 3 it has been possible to calculate the percentage decrease in consumption at different power levels for each of the high octane fuels, compared with the 70 octane fuel, and for the benzole mixture compared with the high octane fuel. These results are shown in Figs. 4 and 5.

Conclusions

(1) An A.40 engine operating at full throttle develops approximately the same power when run on a 70 octane fuel, on a high octane petrol or on a benzole mixture of similar octane number if the ignition setting of T.D.C. recommended by the maker of the engine is used.

(2) If the ignition is adjusted to give maximum power at full throttle on the 70 octane fuel, greater power is developed when the engine is run at the same ignition setting but with a higher octane fuel.

(3) The greatest increase in power is obtained when the engine is run on high octane fuels with the ignition adjusted for maximum performance.

(4) The effect of the greater power output with the higher octane fuels is to enable a given power output to be obtained with a smaller throttle

opening and thus with a lower fuel

consumption.

(5) With the Zenith carburettor the consumption with all fuels at the lower power outputs is less than with the S.U. carburettor, although at full throttle the S.U. carburettor enables a greater power output to be obtained. The fuel consumption for the S.U. carburettor at a power output equal to the best obtainable with the 80 octane fuels used in conjunction with the Zenith carburettor is lower than with the Zenith.

(6) The percentage saving in fuel when the Zenith carburettor is used is

about 12 per cent for the benzole mixture, and rises from 5 to 8 per cent, according to power output, for the 80 octane petrol in comparison with the 70 octane petrol. The consumption for the benzole mixture as compared with that of the 80 octane petrol shows a saving falling from 8 per cent to 5 per cent according to power output.

(7) With the S.U. carburettor the savings are generally less, rising according to power output, from 6-5 to 9 per cent for the benzole mixture and from 4 to 5 per cent for the 80 octane petrol compared with 70 octane petrol. The consumption with this carburettor

is only from 2 to 5 per cent lower for the benzole mixture than for the 80 octane petrol.

(8) From these results it is seen that the advantage of an 80 octane petrol compared with a 70 octane fuel, or of a benzole mixture compared with either a 70 octane petrol or a petrol of equal octane number, is a function of ignition setting, and probably depends also in some measure upon carburettor setting and design.

Further experiments are necessary to determine the effect of benzole content and carburettor settings on the air-fuel ratio supplied to an engine.

PERMANENT MAGNETS

An Important Development in Powder Metallurgy

FOR more than twenty years The General Electric Co. Ltd. has produced iron dust components for electrical work, and now this organization has developed a technique for making magnets—equal to any made from the finest cobalt steel—from microscopically fine iron dust produced by a special process. These new magnets will be known commercially as Gacalloy magnets, and will be produced by The Salford Electrical Instrument Co. Ltd., a G.E.C. subsidiary.

The development work on iron powder inductance cores for many different applications showed the way to the production of magnets as well as of magnetic materials. However, experiments showed that the relatively coarse powder that was suitable for the core of an inductance was not at all suitable for a permanent magnet. On the contrary, a fine powder was needed, and in this context the word "fine" is almost an understatement. In fact,

the particles used are only 100 times the diameter of an atom of iron and 1/1000th the diameter of the finest radio iron powder hitherto manufactured.

To produce the minute particles of pure iron was in itself difficult, but in addition there were certain problems of handling to be solved. Many fine metallic powders when loose are pyrophoric, that is, they ignite on coming into contact with air. Therefore method of handling had to be evolved that would remove this danger. This problem was solved almost two years ago but experimental production was continued until the Company was satisfied that the magnets produced by this completely new process were not only equal in performance to conventional magnets but were also indeed permanent. Exhaustive tests have proved conclusively that all the requirements of a good magnet are met by the new process.

The different steels used hitherto for magnets have all had one property in common; they are extremely hard. As a result fabrication methods are limited to casting and grinding, and are therefore expensive and relatively slow. By contrast Gecalloy micropowder magnets can be made in a great variety of shapes by the use of power presses and special press tools. There is virtually no restriction to the shapes that can be produced.

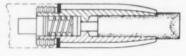
Another factor of importance is that strength for strength a Gecalloy magnet weighs only approximately half as much as a conventional steel magnet. Furthermore, the development of these magnets opens the way to major price reductions when large-scale manufacture is attained. In addition, the fact that every particle can be insulated from every other particle means that the magnet is an insulator, and there are no eddy currents in a Gecalloy magnet used in an inductive device.

FLO-MASTER FOUNTNBRUSH

NEW brush with a fountain pen A NEW brush with a foundation been type reservoir has recently been cushman introduced in this country by Cushman and Denison Co., Ltd., of 124, Vassall Road, London, S.W.9. The pocket size version may be carried around in the same way as a fountain pen; but a larger model, the King size, is made for use at a work bench. There must be innumerable applications in industry for these units, such as for marking stores or rectifications needed on inspected components. They would also be useful in packing and despatch departments, and advertising artists might be interested. A noteworthy feature of this unit is that relatively little skill is required, by comparison with that needed for brush work, for ordinary marking operations.

The manufacturers claim that the Fountnbrush will make a permanent

mark in the shortest possible time on any surface including glass, plastics, leather, waxed paper, cellulose film, timber, steel, chromium and ordinary paper. The ink dries almost instantaneously when applied to an absorbent surface such as timber or paper, and after a relatively short time on non-



The nib holder and valve

absorbent surfaces. It may be obtained in containers of six different sizes, from 2 oz to 1 gallon, and in ten different colours. All colours, except white and silver, which are semi-opaque, are available in two grades: transparent for general purposes, and semi-opaque for heavy marking, especially on metal glass and dark surfaces, etc.

Both the pocket size and the King size brushes are similar in construction. They have a hexagon headed, screwedon filler cap at one end, and a nib holder and valve at the other. The nibs are of a good quality hard felt and a large range of sizes and shapes are available. They are protected when not in use by a cap which in the pocket size has a clip on it, and which in the King size may be screwed to the bench to form a permanent stand for the unit. During use, if pressure is applied to the nib it slides up in its holder and lifts a spring loaded valve off its seat to allow ink to flow down from the reservoir in the body of the brush.

THE PERKINS P3 (TA) ENGINE

A Diesel Power Unit Incorporating Many Components Common to the P4 and P6 Engines

CINCE the war there has been a Steadily increasing demand for light and medium diesel engines. This demand arises not only because of the widespread trend towards mechanization on farms, but also because of the flat torque characteristics and operational economy of compression ignition engines. These features make them better suited than petrol engines for both tractors and commercial vehicles. The movement towards the adoption of diesel units has been further accelerated in some countries by unduly heavy taxation on fuels for petrol engines.

The Perkins P3 was initially designed as a conversion pack for the Ferguson T.E.-D20, V.O. engined tractor, but it is also suitable for the T.E.-20 (Continental) and the T.E.-A20 standard petrol engined unit. Later it was incorporated in other tractors and in the Trojan 15/20 goods carrying chassis. The three cylinder layout was adopted because it so happened that half of the six cylinder engine developed the power needed for the particular application for which it was designed, and space require-ments were conveniently met. Moreover, the rationalization that was possible with this arrangement permitted considerable economies in production.

Nearly all the parts subject to wear

SPECIFICATION

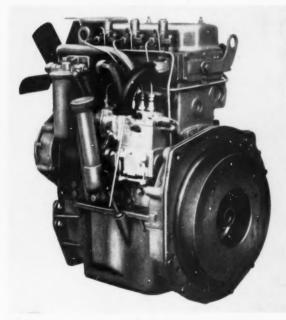
Three cylinders. Bore and stroke 3.5 in (89 mm) × 5 in (127 mm). Swept volume 144 in³ (2,360 cm³). Maximum b.h.p. 34 at 2,200 r.p.m. Maximum b.m.e.p. and torque respectively 92 lb/in2 o.m.e.p. and corque respectively y2 lorin-and 88 lb-ft at 1,300 r.p.m. Compres-sion ratio 16·5:1. Forged four bearing crankshaft with bolted-on balance weights. Overhead valves; rockers directly operated by tappets on high directly operated by tappets on high camshaft. Injector pump C.A.V. type BPE 3A 600 310/3S 6230. Injector nozzles: C.A.V. type BDL 110 S6036 atomizer in a BKB35S87 holder. Combustion chamber: Perkins patent "Aeroflow", swirl chamber. Fuel lift pump: C.A.V. type BFP/K22 P68. Fuel filters: C.A.V. type BFASP2 and Tecalemit FD 2161.

are common to the P3, P4 and P6 engines, so that servicing is appreciably simplified. The pistons, connecting rods, bearings, valves and timing gear, pulleys, etc., are all the same for the whole range. Only components affected by the change in length are different

The firing order is 1.2.3. From the data given in the specification panel, it can be seen that the bore: stroke ratio is 0.7:1. The connecting rod length: stroke ratio is 1.8:1. At 2,200 r.p.m.,

when maximum b.h.p. is developed, the mean piston speed is 1,830 ft/min. The maximum brake mean effective pressure is 92 lb/in2 at 1,300 r.p.m., and the i.m.e.p. at the same engine speed is 115 lb/in². From this, it follows that the mechanical efficiency is 80 per cent. The b.h.p. per square inch piston area is 1.18, and in terms of b.h.p. per litre a figure of 14.4 is obtained. The minimum brake specific fuel consumption is 0.397 pint/b.h.p/ hr. Without the dynamo, starter, flywheel and its housing, the engine weighs 630 lb dry, and its overall dimensions are 30 in high, by 22½ in wide by 231 in long. These dimensions are for an engine without the air filter and flywheel.

Finding space for all the components on each side of such a short engine obviously has not been easy. The dynamo is pivot mounted on the righthand side, where it partly overhangs the starter. On the left-hand side is the injector pump which is shorter than usual, since it contains only three elements. An oil filter is mounted on a bracket bolted to the cylinder head above the injector and oil pump drive. Below the drive, a cast iron oil filler tube, to which is welded a breather pipe, is bolted to the side of the crank-The inlet pipe is on the righthand side of the head, and the exhaust





there is not much room for the components on this engine

Despite the short length of the three element injection pump, The Perkins P3 is virtually half of the P6, and most components subject to wear are common to both these models and the P4

THE THREE CYLINDER PERKINS P3 T.A. DIESEL ENGINE Bore and stroke 3½ in . 5 in. Swept volume 2,360 cm³ 00 0 0 H A.

on the left. The injectors are bolted to the top face of the head above the exhaust manifold. At the front, there is a six-bladed fan with the usual triangulated V-belt drive for the dynamo.

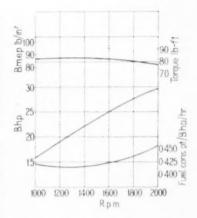
The cylinder block and crankcase

An integral cylinder block and crankcase of nickel or chromium cast iron is employed. It is of exceptionally stiff construction, having two transverse webs to support the intermediate journal bearings, and both the webs and walls extend down to the level of the axis of the crankshaft. At the rear, a face is machined to carry the flywheel housing which is located by two dowels. The joint is sealed by a 0.010 in thick paper washer. The timing case is assembled, together with a ½ in Klingerit washer, to a machined face on the front end.

Dry liners with a wall thickness of ½ in are fitted in housings, which have a wall thickness of about ½ in. Adjacent cylinder liner housings are joined together so that there is no water jacket space between them. At each junction, the metal thickness is approximately ½ in. The liners are of

cast iron with a Brinell hardness number of 225-255. They are fine bored and hone finished. At the upper end, the outer periphery of each liner is flanged. Immediately below the flange is an undercut that eliminates the need for continually re-dressing the edge of the grinding wheel during production. The flange, which is only $\frac{1}{16}$ in wide, seats on a shoulder in the housing, and it is held down by the cylinder head and gasket.

The main journal bearing caps are also of cast iron. They are held down by 16 in diameter En 18T set bolts around which are dowel tubes for location purposes. The front and rear bearing caps are each shouldered and fitted with a semi-circular cork washer, against which the sump bears to complete the oil seal, and their set bolts are outside the sump. This arrangement makes the sump rather short, but it was adopted in the interests of rationalization. Locking is effected by means of tab washers. Both end caps incorporate a drainage passage, at the front the passage is about 16 in diameter to return the lubricant from the timing case to the sump, and at



Engine performance curves obtained with dynamo on charge, and with a 15 in diameter, 6-bladed fan fitted

the rear it is \(\frac{1}{6}\) in diameter to drain the casting surrounding the crankshaft rear oil return and thrower.

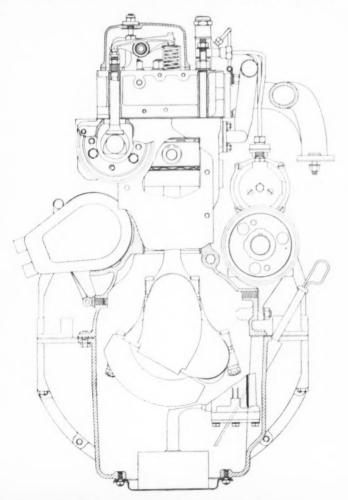
Crankshaft, connecting rods and pistons

An En 19T crankshaft is fitted. Its four main journals are Tocco hardened to V.P.N. 550-600, and the remainder of the shaft has a Brinell hardness number of 248-302. The three crank throws are spaced 120 deg apart so that the crankshaft is virtually half that of the six cylinder engine. With this arrangement, there are no primary and secondary unbalance forces, but there are both primary and secondary couples. The rotational balance is effected by two cast iron counter weights bolted to the end crank webs.

The main journals are $2\frac{1}{4}$ in in diameter and are carried in steel backed copper lead bearings of Glacier manufacture. Both the intermediate bearings are 1 in long; the length of the front bearing is $1\frac{1}{16}$ in and that of the rear $1\frac{1}{8}$ in. On all bearings, the radial clearance is 0-00175-0-0025 in. Location of each shell is by means of a dowel in the connecting rod half.

Between the front of the front web and the back of the rear web the length of the crankshaft is 11½ in. The thickness of the front and intermediate crank webs is $\frac{\pi}{8}$ in, while that of the rear one is $\frac{\pi}{4}$ in. All are $3\frac{\pi}{8}$ in wide measured across the crank pin axes. The diameter of the crank pins is $2\frac{\pi}{4}$ in.

Axial location of the crankshaft is effected at the lower half of the rear bearing shell, which is flanged on each side of the cap to bear against the crank web on one side and against the oil thrower ring on the other. This thrower ring is integral with the crankshaft, and immediately to the rear of it is an oil return scroll which works in a twopiece aluminium die-casting. The upper portion of the die-casting is bolted to the crankcase and the lower portion to the bearing cap, and the halves are held together by a 4 in diameter bolt and split pinned nut vertically positioned on each side.



The tappets are carried in the cylinder head

Four ½ in diameter set bolts secure the B.S.1452 grade 17 cast iron flywheel to a flange on the tail end of the crankshaft. This flange is spigoted into the front face of the flywheel, and further positive location is effected by a dowel. Support for the front end of the clutch shaft is afforded by the ball bearing carried in a cast housing that is spigoted into the rear of the flywheel and secured by two set bolts. This arrangement is probably necessary

because of the thickness of the flywheel, otherwise the clutch shaft, were it carried in the tail end of the crankshaft, would be unduly long and slender. The flywheel is 14½ in diameter by 3½ in thick overall and weighs 124 lb. An En 8 starter ring gear with 120 teeth is

spigoted on to the flywheel and is secured by four fe in diameter set screws.

The En 24U connecting rods are oil quenched at 840 deg C and tempered to Brinell hardness number 269-321. Their centre-to-centre length is 9 in and the big end bearings are split at 90 deg to the axis of the rod. Location of the caps is effected simply by the fit of the ¼ in diameter En 110V bolts, the slotted nuts for which are locked by split pins. Vandervell thin wall, lead bronze lined, indium flashed, big end bearings are employed. The bearings are 1¼ in long, and the radial clearance is 0.0016 in-0.0021 in.

A lead bronze lined bush is pressed into the small end of the connecting rod. In it is carried a 1½ in diameter gudgeon pin drilled out ½ in diameter; the running clearance is 0.001 in on the diameter. The gudgeon pin is made of case hardened En 353 or En 354, and

is an interference fit in the piston bosses which are $\frac{7}{6}$ in long. Positive axial location is afforded by Seeger circlips in the grooves in the piston bosses.

Low expansion aluminium alloy plain pistons are fitted. Their crowns are flat except for a small cavity which is positioned in line with the transfer port of the pre-combustion chamber. Each has three compression rings and two oil control rings. The top two

CAMSHAFT PERFORMANCE DATA AT 2,200 R.P.M.

Maximum positive acceleration of tappet (flank)	2,130 ft/sec ²
Maximum negative acceleration of tappet (nose)	792 ft/sec ²
Maximum tappet velocity	5·12 ft/sec
Lift at camshaft	0·312 in
Nominal period of cam	118 deg

compression rings are of plain rectangular cross section and their principal dimensions are: gap 0.009-0.013 in, side clearance 0.002-0.004 in, face width in, radial thickness 0.143-0.135 in. A Cords laminated lower compression ring is employed. Each of the three laminations is 3/2 in thick, and the radial thickness is 0.127-0.123 in. All grooves in the piston are 0.160 in deep. A Duoflex oil control ring is fitted below the gudgeon pin and a slotted ring above. The face width of both is 1 in while the side clearance and radial thickness of the slotted ring are respectively 0.002-0.004 in and 0-135-0-137 in. The radial thickness of the Duoflex ring when in position is 0.160 in. Again, the depth of both grooves is 0.160 in.

Timing gear, camshaft and valve gear

A three-piece LM4M aluminium die-casting encloses the timing gears.

One piece is the dished casing bolted to the front of the crankcase, the second is its front cover, and the third is a small closing plate bolted on below the level of the sump face joint. From the point of view of weight saving this arrangement represents an appreciable gain as compared with the more common practice of casting the casing integral with the cylinder block. A cast, rather than a pressed steel, front cover is necessary to provide adequate stiffness for supporting the water pump which is

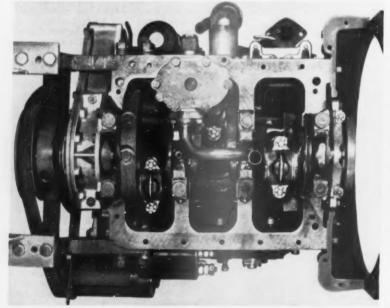
stiffness for supporting the water pump which is mounted on it. No doubt this has another incidental advantage in that it reduces the noise transmitted from the timing gears and chain drive.

A three - strand roller chain of 3 in pitch transmits the drive to the camshaft and injector and oil pump drives. Chain tension is maintained by a Renold and Coventry Autotensioner, type 622564, bolted to the front wall of the crankcase. The En 8Q driving sprocket together with the 51 in diameter B.S.1452 grade 14 fan belt pulley is carried on the 11 in diameter front extension of the crankshaft. A single plain key, 23 in long, drives both wheels. These wheels are pulled against the shoulder formed by the front main bearing of the crankshaft by the 7 in diameter special bolt incorporating the dogs for the starting handle. An Angus MIS 20 synthetic rubber oil seal is housed in the front cover and bears on the boss of the fan belt pulley. A dished oil thrower ring clamped between the pulley and the sprocket partly enshrouds the seal and its housing.

The camshaft is carried in a cell high on the right-hand side of the cylinder block, so that push rods are not needed. This arrangement is a Perkins patent. Several advantages are claimed for it: the elimination of push rods in weight and cost; economizes because of the reduced weight of the moving parts lighter valve springs may be employed; the system is more rigid and therefore less susceptible to troublesome vibrations; variations in tappet clearance due to thermal expansion are reduced; and the tappet adjusting screws may be carried in the tappet instead of in the rocker. locknut for the adjusting screw is large enough to prevent the tappet from falling out when the cylinder head is removed.

A close grained cast iron half speed wheel is secured to a flange near the front end of the camshaft by three he in diameter bolts. There are eight bolt holes in the sprocket and three in the camshaft flange, and with 40 teeth on the sprocket, the relative position of the two components may be adjusted in steps of 1½ deg.

Three bearings in transverse webs in the cell carry the $1\frac{3}{12}$ in diameter Monikrom camshaft. The front bearing is 1-87 in diameter by $1\frac{1}{2}$ in long, the second is 1-86 in by 1-0 in and the third is 1-84 in by $1\frac{3}{2}$ in. No bushes



Bolted on balance weights are fitted to the crankshaft

VALVE DATA

are employed. Axial float is controlled by an arched leaf spring. One end of this spring is riveted to the front cover, and its centre bears against the end of the camshaft, pushing it to the rear until a shoulder behind the flange carrying the half speed wheel bears against the front face of the cylinder block. A projection raised on each side of the front cover prevents the free end of the spring from turning about the riveted end.

Close grained chromium iron, mushroom type tappets, the stems of which are $\frac{3}{8}$ in diameter, are carried directly in the cylinder head. The radial clearance between the head and the tappet is 0-0005-0-00175 in. A case hardened En 32 tappet adjusting screw and a lock nut are carried in the upper end of each tappet. The overall length between the adjusting screw and the head is $5\frac{3}{16}$ in.

and the head is 5 ½ in.

Four LM4M or LM6M die-cast pedestals support the case hardened En 32 rocker shaft which is \$\frac{1}{6}\$ in outside diameter by \$\frac{1}{6}\$ in inside diameter. The shaft is located

inside diameter. The shaft is located axially by a circlip at each end, and is prevented from rotating by the clamping action of the pedestal holding down studs in the split bosses that carry it. The En 9S rockers have induction hardened end pads. Pressed into their bores are lead bronze lined, steel bushes. Each rocker is constrained against its adjacent pedestal by a compression spring round the shaft.

The valves are positioned in line in the plane of the cylinder axes. The distance between the axes of each pair of valves is 1½ in. Dimensions and other details of the valves are given in the table. Two springs are employed on each valve; they are retained by a washer and split collets. The lower end of the inner spring seats on a stepped washer around the valve guide while the outer spring seats on the cylinder head and is located by the stepped washer.

The valve guides are 25 in long by in outside diameter. They are interchangeable. Their upper ends are tapered externally and shouldered to bear on the top of the cylinder block. They are also tapered externally at their lower ends where they project in from their bosses. For a length of in from the end they are counterbored to a diameter of 3 in. counterbore is intended to shield from direct contact with the exhaust gas stream that part of the stem adjacent to the guide. It also provides a space in which any carbon that may be deposited is confined without detriment to the bearing portion of the guide. The valve seats are cut directly in the cylinder head. When in the

	Inlet	Exhaust	
Material	, DTD 13B		
Head diameter	1 % in	1 5 in	
Throat diameter	1 ¾ in	1 in	
Stem diameter	in in		
Seat angle	90 de	g included	
Spring rate:			
inner	42.5		
outer	77.5	lb in	
Spring length free:			
inner		-1·405 in	
outer	1.803	−1·783 in	
Spring length installed:			
inner	1 % ir	1	
outer	1½ in		
Surge frequency:			
inner		0 c.p.m.	
outer	18,00	0 c.p.m.	
No. of coils:			
inner	71		
outer	71		
Coil diameter:			
inner	∦ in		
outer	1 16 ir	1	
Wire gauge:			
inner	13 s.v		
outer	0.136		
Valve lift	0.350		
Rocker ratio	1.15:1		
Valve crash speed		r.p.m.	
Valve guide length	2∮ in		
Tappet clearance, hot	0·010		
Valve opens		46 deg B.B.D.C.	
Valve closes	43 deg A.B.D.C.	10 deg A.T.D.C.	

closed position, the valve heads are recessed approximately 1/4 in above the lower face of the cylinder head. Further details of valve and seat dimensions, etc., are given in the table.

Cylinder head, manifold and injection equipment

Fourteen is in diameter En 18T studs are employed to hold down the cylinder head. Six of them are of extra

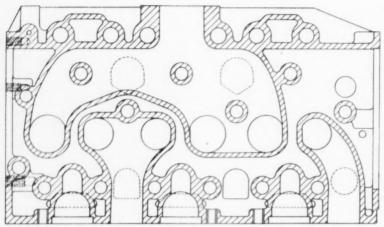
length to hold down the injectors. The overall dimensions of the head casting are: 3 in deep by $8\frac{1}{8}$ in long. The porting arrangement is somewhat unusual, since it is half of the six cylinder layout. As can be seen from the illustration, two of the exhaust ports are siamesed and the third is separate. All the inlet ports open into a common chamber to which the LM4M induction pipe is bolted.

The combustion chambers are of the Perkins patent "Aeroflow" design. They are more or less spherical with an elbow shaped transfer passage communicating with the cylinder. Each combustion chamber is cored and machined in the head and closed by an En 5B boltedon portion. A copper sealing washer is fitted between the head and the closing piece. A two-spray injector is employed. It discharges into the elbow of the transfer passage, and directs one spray of fuel into the swirl chamber and a second down

the transfer port into the cylinder, thus combining the flexibility obtained from a swirl-type combustion chamber with the efficiency of the direct injection type head. For cold starting, a Kigass pump is used to pump diesel fuel through a sprayer nozzle in the induction pipe. Part of the spray is directed on to a hot plug. The fuel is ignited and burning mixture is drawn into the cylinders to



A pipe line carries oil to the centre of the rocker shaft



A section through the cylinder head, showing the valve port arrangement

warm the induction system and combustion chambers.

In applications in which the fuel tank is lower than the injector pump, a C.A.V. BFP/K 22 P 68 fuel lift pump is employed. The two fuel filters are the Tecalemit FD.2161 and C.A.V. BFA 5P2 types in tandem. A C.A.V. BPE 3A 600 310/3S 6230 injection pump is mounted on a bracket on the nearside of the cylinder block. It is used in conjunction with a C.A.V. BEP/MZ80A/100 pneumatic governor. The pump delivers the fuel at a nominal working pressure of 120 atmospheres to the C.A.V. BDL 110 S6036 atomisers in BKB 35 S87 holders. The injection pump spill timing is 30 deg before top dead centre

Lubrication and injection pump drive

The lubrication and injection pump drive is carried in an aluminium casting bolted on the left-hand side to the rear of the timing case. The close grained, cast iron driving sprocket is pressed on and keyed to the 14 in diameter En3 spindle. The spindle is carried in two phosphor bronze bushes spaced 21 in apart. A flange on the forward end of the 14 in long front bush takes the thrust. The rear bush is 11 in long and is pressure fed with oil from the head of the Tecalemit filter and an oil weir at the front end of the casing maintains the level to lubricate the front bush. Behind the rear bush is a case hardened En 32 thrust washer interposed between the housing and the spiral gear which is keyed on the I in diameter rear end of the spindle to engage with the gear on top of the oil pump drive spindle. The driving gear is retained in position between a shoulder on the spindle and a Seeger circlip at the back. To the rear of the gear the spindle is further stepped down in diameter to in where it is passed through a lip type seal housed in a bolted on end cover. The C.A.V. coupling for the injector pump is keyed on to the tail end; it incorporates a vernier device for timing adjustment.

Oil pump and lubrication system

A two-piece, En 32 pump drive spindle is employed. Its upper portion is housed partly in the drive casing just described and partly in the crankcase, while its lower portion is in the oil pump body. The top end of the upper portion is reduced to a in diameter and pump bedy. is carried in a ball bearing in a housing cap bolted to a face machined on top the drive casing. Immediately below the bearing the spindle is 1 in diameter and carries a pressed on phosphor bronze, spiral driven gear. The drive is through a Woodruff key. Below the shoulder against which the gear is located, the shaft is & in diameter and ground for a length of I in, where it is carried in a phosphor bronze lined, steel bush. This bush is housed in a boss in the drive casing; its upper end is flanged and takes the thrust from the driven gear.

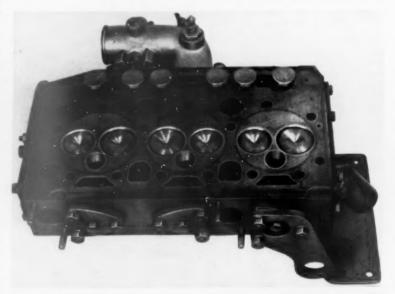
Below the bearing, the spindle is

turned down to re in diameter. It is surrounded by a sleeve, the upper end of which is belled to seat on a rubber ring round the end of the bush projecting below the drive casing. The lower end of the sleeve extends into the spindle housing in the crankcase. A compression spring is carried between the belled upper end of the tube and a conical washer fitted loosely round the lower end. The washer seats on a rubber ring which is thereby held firmly round the tube and at the same time pressed against the top of the crankcase to form an effective oil

The bottom end of the upper portion of the spindle is dogged and engages in a slot in the top end of the lower portion. This dog and slot coupling is just below the level of the sump face joint, in a phosphor bronze bush, 7 in long, in a boss at the top of the D.T.D. 424 pump body casting. Another phosphor bronze bush, also in long, spaced 3½ in below the first one, carries the lower end of the spindle on which is pressed the pump

driving gear.

Both the driving and driven gears are of En 6. They are ½ in long by 11% in diameter. These gears are shorter than on most designs in order that most of the pump components may be common to the 3, 4 and 6 cylinder Perkins engines. For the same reason, that portion of the casing which surrounds the gears is a separate piece. This arrangement may have another incidental advantage in that it can be produced by through-milling instead of end-milling. The pump body, therefore, consists of three pieces: the upper portion, the lower end of which is machined flat; the gear chamber; and the flat end-cover which, together with the gear chamber, is bolted on to the upper portion. The spindle for the driven gear is 1 in diameter and is



Each swirl chamber is closed by a cap held by three studs on the side of the cylinder head

of cast iron. There is a spiral groove round the lower part of its periphery to assist lubrication. This spindle is pressed into the pump casing and located against rotation by a peg engaging in a slot in its upper end.

Because of its short length, the cast aluminium sump is deeper than most designs; it holds 1½ gallons. The oil pump is about 3 in above the oil level. Oil is drawn through a pipe from a gauze strainer mounted on a pressed steel cover plate bolted to an aperture in the base of the sump. The outlet from the pump is a 1 in diameter duct drilled in the pump body. This duct passes the oil vertically up to a chamber formed by an aluminium casting bolted over the relief valve fitted in the side of the crankcase. The blow-off pressure is 60 lb/in². From the chamber a pipe carries the oil up to the Tecalemit FA 2689 filter. An external pipe from the filter head takes the lubricant to the gallery drilled longitudinally in the crankcase. Thence the oil passes through ducts to the three main journals; passages drilled in the crankshaft distribute it to the big The small ends are splash lubricated. As has already been mentioned, another pipe carries oil from the head of the filter to the rear bearing of the pump drive.

A third pipe communicates between the head of the filter and the centre cam haft bearing. Oil issuing from this bearing fills the camshaft cell to the height of an oil level weir over which it passes into a drain channel incorporated in the outer wall of the cell. This channel is open at its front end and discharges oil on to the timing chain and sprockets.

Drillings in the camshaft centre bearing are arranged to deliver an intermittent feed up through a passage to the cylinder head. A union is screwed into the upper end of this passage and oil is piped from it to a boss fitted round the centre of the rocker shaft. From here it passes into the shaft, and the rockers are lubricated in the usual manner through the radial holes. A vertical drilling through the rockers and their bushes allows oil to pass out on top, so that it may run down to lubricate the tappet ends.

Water pump and cooling system

The water pump is mounted on the front cover of the timing case. It is driven at 1-3 times engine speed by a fan belt, $\frac{2}{8}$ in wide by $\frac{5}{8}$ in thick with a V-angle of 28 deg. The pump body is of LM4M cast aluminium. A $2\frac{1}{4}$ in diameter B.S. 1452 grade 12 cast iron impeller is employed. It is pressed on to the $\frac{1}{4}$ in diameter rear end of the spindle, and holes are drilled and tapped in its boss to accommodate an extractor.

A spring loaded moulded rubber water seal is employed. At the rear a carbon thrust ring bears against the rotor boss, and at the front the seal is retained in position by a washer seating on a shoulder in the housing. Forward of the water seal the spindle diameter is $\frac{1}{8}$ in, 1/6 in apart. Two grooves are machined around it to form the inner races of the two row, tubular type ball bearing.

Between the rear of the bearing and the water seal, an L-section steel thrower ring is pressed on the spindle and is housed in a drainage space. In front of the bearing the B.S. 1452 grade 12 cast iron pulley is pressed on to the spindle. The six-bladed fabricated pressed steel fan is bolted and spigoted on the pulley. The whole assembly, comprising the pump rotor, spindle, water seal, thrower, bearing and pulley is located by a grub screw in a tapped hole on top of the nose of the body This screw engages in a hole casting. in the tubular outer race of the bearing. The coolant circulation is from the pump through a pipe to a connection near the rear of the left-hand side of the cylinder block. It then flows through ducts into the head, on the front end of which is a water outlet.

Electrical equipment

Twelve-volt electrical equipment is employed. The battery capacity is 63 amp-hr on a 10 hr rate. It is served by a Lucas C39P/2 dynamo used in conjunction with an RF96/2 L13 voltage regulator. A C.A.V. BS512K66 axial type starter is fitted and the pinion has 11 teeth to engage with the 122 teeth on the starter ring gear.

ALLOY STEELS

Conservation of Alloys in Steels for Truck Axle Parts

IN a paper entitled "Alloy Conservation in Steels for Truck Axle Parts" by T. A. Frischman, in Eaton Forum, September 1952, the experiences are given of the Axle Division of the Eaton Manufacturing Company in the adoption of substitute steels as a result of a conservation order by the U.S. Government in 1951. This order limited the use of the strategic alloying elements Ni and Mo. Conservation methods included reduction of alloy content, replacement by non-strategic alloying elements, and the use of boron steels.

Tables showing the specifications and analyses of the regular and replacement steels, together with some of the parts for which they are used, are given. The carburizing types are now limited to a maximum of 0-60 per cent Ni and 0-15 per cent Mo, while in the through-hardening types, the main differences are in the reduced Mo content of the TS 4100 series steels, and the reduction of both Ni and Mo in the others. The Mo reduction in both instances was compensated for by an increase in Mn and Cr content and, in the Cr-Ni-Mo types, the reduction in Ni and Mo was made possible by the addition of B.

The choice of a suitable steel is influenced by machinability, hardenability requirements, and response to heat treatment, as well as by availability. In sections of less than 3 in, the boron through-hardening steels are comparable as regards physical properties with the regular steels they have replaced. It is stated that the standard Jominy end-quench test is a reliable guide for these comparisons.

Statistical plots of the hardenability of various heats of through-hardening steels showed that the emergency steels TS 86B45, TS 4150 and TS 81B45 would form suitable substitutes for the 4340H, 4150H and 6150H grades previously used for axle shafts. In another application, the boron steel TS 81B45 was selected to replace 8645 for a high-speed clutch plate. Apart from the equal hardness and even greater toughness of the replacement steel, it had the further advantage of lower annealed hardness, so giving easier machining. By contrast, the boron carburizing steels are more difficult to work with. Distortion control and bore contraction are unpredictable and make the achievement of proper tooth contact in spiral bevel and hypoid gears a problem. The

effects of boron on machinability and hardenability are discussed, and a method of measuring the case-hardenability on end-quench specimens is described. M.I.R.A. Abstract No. 6148.

The Queen's Air Forces

THOSE of our readers who are interested in aviation may like to know that our associate journal, Flight, is producing a special Coronation Number, entitled "The Queen's Air Forces," on the 29th May. This is a souvenir issue and it will review the work and equipment of the R.A.F., Naval Aviation and Commonwealth Air Forces to-day, and recall the most important events in British aviation during the lifetime of H.M. Queen Elizabeth II.

A prominent feature will be air-toair photographs in full colour giving a first-hand account, specially reported by Flight during a tour of Middle and Far East units, of the work of squadrons operating in Korea and Malaya.

Copies of the issue can be obtained from all newsagents, price 1s. 6d. This is sixpence more than the usual price.

LIGHT ALLOY BODYWORK

A Review of Mass Production Practice in France

XAMPLES of the use of aluminium in touring car bodywork go back a good many years. Perhaps the earliest example is a 1907 de Dion with a body of aluminium sheet Between 1922 and 1924 several builders supplied bodywork, often of great luxury, completely in aluminium, and in Revue de l'Aluminium for October 1924 there are a number of illustrations of well-known makes with aluminium bodies. In 1938 the situation in France was: Voisin was producing aluminium bodies; Talbot was building in small quantities a car with a welded aluminium body; the body building firm of Million-Guiet was using the construction principle patented by Viscaya for pro-ducing thousands of aluminium tourer bodies; and Citroen was fitting Million-Guiet bodies to a special model with an all-aluminium engine.

Despite all these developments, the problems of mass production for aluminium bodies had not been seriously attacked. So long as price was

attacked. So long as price was not an important factor, there was no difficulty in producing these bodies in pure aluminium of a fair thickness, formed more or less by hand and assembled by riveting, fasteners and a certain amount of torch welding. Nevertheless, these cars proved that aluminium would give satisfactory

service in this application and s h o w e d the p o s s i b i l i t y of improved performance through the lightening of the vehicle.

During the period 1932-1939, the basic problems were investigated by the large automobile manufacturers—such as Citroen and Peugeot—and by deep drawing specialists such as Chausson. The results of these activities were not spectacular, but they did demonstrate the following:—

(1) The impossibility of using the standard



Fig. 1. Bonnet of the Dyna Panhard

alloys (99.5 Al, Al-Mn, Al-Mg-Si, cast) for reasons either of mechanical strength or of complication of heat-treatment.

(2) The necessity of inducing spotwelding machine makers, Sciaky and Langeupin, to produce stationary machines or pinch welders for the assembly of light metal parts.
(3) The need for specialists to consider the problems of fillers and painting which had

hitherto not been solved.

(4) The advantages of lightweight bodywork and the cost per pound of weight saved. Between 1937 and 1939 Citroen had developed a front-wheel drive 11 h.p. car entirely of light metal, which weighed 660 lb less than the ordinary steel car and had quite a remarkable performance. At the same time, Citroen were preparing for the mass production of a small car — the present 2 h.p. — which at that date was intended to be entirely of aluminium. At pre-war values the additional cost per Kg of weight saved was in the order of eight francs; to-day it is 160 francs, that is the additional cost per pound of weight saved was 1s 4d before the war and is now 3s 4d.

The period of the occupation of France was not one of inactivity. It was, in fact, during that period that L'Aluminium Francais and

J. A. Gregoire decided to co-operate in planning and developing a small car to meet post-war requirements. Eventually, however, it was the Société Panhard which decided to go into production, at the rate of 50-100 per day, with a car having bodywork completely in light metal and produced by

the most modern procedures.

Choice of alloy

When the decision was made to produce this car, important two points had to be considered :.. (a) the choice of the alloy, and (b) the thickness of the metal for the deep-drawn parts of the bodywork. Neither pure aluminium nor manganese alloy was suitable. They are excellent materials for deep-drawing, but their low strengths would have entailed the

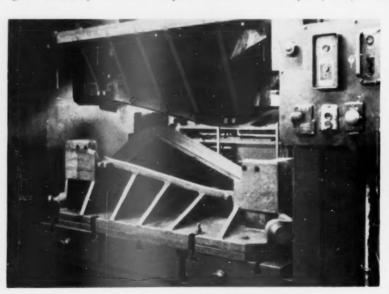


Fig. 2. Drawing tool for the Dyna Panhard half-bonnet

Based on a paper by J. Baron, Director of Technical Services L'Aluminium Français.

use of gauge thicknesses that were incompatible with the stipulated selling price and lightness. It was therefore decided to use a magnesium alloy, Al-Mg3, that had given the best results during the pre-war investigations.

There was thus only a single light alloy, whereas for steel bodies there is a whole range of steels for the different deep-drawing problems. Furthermore, there was little practical experience on the possible effects of rolling and draw-There were also ing on the sheets. other handicaps. For example, the form of the bodywork was still completely influenced by the use of steel and the deep-drawing characteristics of that metal. Furthermore, because the tool designers lacked practical experience in drawing light metal, the press tools were made as if they had to draw steel. Nevertheless, the problems were all solved eventually, and now the daily production problems are no more serious than would be met with in the production of steel bodywork.

From the outset a normal composition A-G3 alloy was used. It was made from 99.5 per cent pure aluminium with the following additional elements: -

2.5-3.7 per cent 0.2-0.5 per cent Mg Mn Fe Si Less than 0.5 per cent Less than 0-1 per cent Cu To take the place of 1 mm gauge steel, it was decided after preliminary tests to use 1.2 mm sheet. Normally, the thickness should be determined from the following considerations:—

(a) Overall strength, for passenger

protection in the event of a serious accident. To assess this factor, it is not sufficient to take into account the mechanical strength and the elastic limit of the sheet; these factors must also be considered for the chassis and the bracing, and in fact for the general conception of the car. This was particularly so for the Panhard which has a very rigid chassis and a scuttle in cast Alpax that gives very good protection.

It is also necessary to take into account that a deep-drawing steel with a tensile strength of 24 tons/in2 is to be replaced by a material with a tensile strength of 12.7 tons/in2, and as the stiffness varies directly with the cube the comparative strengths are 24 tons/ in2 for steel and 22 tons/in2 for the light alloy.

(b) Strength under local and elastic deformations must also be considered. As this factor depends upon the elastic modulus and this for light alloys is approximately only one-third that of steel, theoretically the thickness should be increased to approximately 1.4. In point of fact experience shows that 1-2 is enough, since the parts of the bodywork most susceptible to local and elastic deformations are well radiused and offer a resistance to local buckling, which in any case is not governed solely by the gauge of the material and the modulus.

This increase of 20 per cent in thickness is an important economic point, and it is worth while considering whether improvement can be effected, even although it might entail certain constructional difficulties. It should be made clear that within this limit of increase it is possible to deep-draw light aluminium sheet with a tool designed for steel of 20 per cent less thickness. This is a great advantage, since in the present transition stage, no builder would have been willing to make light alloys unless they could also be used for steel if necessary

It is generally taken for granted that a large percentage elongation to rupture is a necessary condition for good drawing characteristics in an alloy, and that, on the other hand, a low Erichsen value is a sign of faulty drawing characteristics. That may be true of steel, but it is only very roughly accurate for light alloys and particularly for Al-Mg3. The elongation of this alloy is almost half that of a good drawing steel and a very pronounced slip of the metal under the pressure plate is inevitable. There is, therefore, an increased tendency for wrinkles to form. Crystalling structure plays a prepon-

derant role in deep-drawing. A large crystal structure gives a metal with high elongation, but a rough surface; and if the crystals are too large, the sheet may show breaks before the operation is completed. Very fine crystals go with a good surface and low elongation. It is therefore neces-sary to determine the permissible crystal size by experiment.

In a remarkable study on the deepdrawing of non-ferrous metals at a London Symposium in 1951, Dudley Jevons made some very important observations on the value of the different tests for determining the

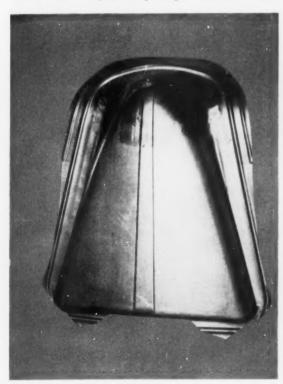


Fig. 3. Bonnet of the Simca 6



Fig. 4. Roof panel for the Dyna Panhard



Fig. 5. The drawing tool for the panel shown in Fig. 4

drawability of aluminium alloy sheets. He stressed the importance of measuring the uniform elongation and not the total elongation in tensile tests. The Erichsen test can supply indirect in-formation of great interest. If the depth of the impression is disregarded, but careful attention is paid to the appearance of the cup and of the break, it is possible from the roughness of the metal at the highest point to deduce the average crystal size and the superficial texture of the drawn piece. Furthermore, from examination of the break, which may be circular, straight or in different directions, it is possible to determine the directional properties of If the metal is likely to the sheet. show the defect known as vermiculation, or Luder lines, this can be verified from the lower part of the cup, which will show very clear indicating

The value of a sheet for drawing may be determined by four ordinary laboratory tests. These, in the order in which they should be applied, are:

 A hardness test (Rockwell or Vickers) which will allow obviously unsuitable sheets to be eliminated.

(2) The Erichsen test to give indications as to anisotropy, crystal size and the appearance the surface will have after the drawing operation.(3) A microscopic examination.

(4) An ordinary tensile test which the metallurgist can interpret from the deep-drawing point of view.

These conclusions, based on experience gained in deep-drawing steel, have proved of value for drawing the Al-Mg3 alloy.

Throughout the experimental period before 1940, the acceptance of Al-Mg3 sheet was on the basis of the Rockwell hardness figure, and one user got as far as establishing a loading schedule. The war prevented the confirmation of this by practical experience until the production of the Panhard car was started in 1945.

Alloy compositions

Many studies have been carried out to determine the alloy composition that will give optimum results. With Al-Mg3 it is advantageous to reduce the iron, silicon and manganese contents. In fact, for difficult parts these con-

stituents should not together exceed 0.15 per cent. This makes it necessary to start with 99.7 per cent aluminium. improve the mechanical properties it would be advantageous to raise the magnesium content from 2.7-3.5 per cent to 5 per cent, an alteration that does not appear to diminish the drawing properties. Unfortunately, the difficulties in rolling this alloy make it too expensive for use in automobile bodywork. Perhaps this economic problem could be solved by the use of more powerful rolling mills. It has established been that in rolling Al-Mg3 alloy, four - high mill produces sheets of better quality than a two-high mill.

Tests have also been carried out to determine whether a nother alloy might not show improvement by:(a) having better mechanical

characteristics (tensile strength close to 19 tons/in²).

(b) possessing drawability at least equal to that of Al-Mg3.

(c) requiring no heat-treatment after delivery to the deep drawer.

(d) being economical in composition and fabrication.

Of the many alloys that were studied, only one held out hope of improvement, A-U2G, of the following composition:—

Copper 2.5-3-0 per cent Magnesium 0.5-0-6 per cent Iron Less than 0.2 per cent Silicon Less than 0.15 per cent Manganese Less than 0.15 per cent After salt bath treatment at 500 deg C, this alloy has the following mechanical

properties:—
Tensile strength 19.7 tons/in²
Elastic limit 11.4 tons/in²
Elongation 23 per cent (17-18 per cent uniform)

Brinell hardness .. 85

Provided the deep-drawing was carried out within eight days of heattreatment, the rear wall of the Panhard could be produced from this alloy in a single draw. The production of the roof for the same car showed up wrinkles which it was difficult to iron out. Unfortunately this alloy ages, and in a manner unfavourable to the drawing characteristics. After six months



Fig. 6. Formation of folds under the blank holder on the Dyna Panhard roof panel



Fig. 7. Front wing of the Panhard in Al-Mg3

the elastic limit goes up by some 1.6 tons/in², while the uniform elongation goes down by 2.5 per cent. Because of this it was not considered to be suitable for replacing Al-Mg3.

Deep drawing technique

The bonnet of the Dyna Panhard, see Fig. 1, was the first part to be produced. In spite of a reduced scale model trial, the maker of the tool considered it was impossible to produce this part in a single operation and preferred to make it in symmetrical halves from the tool shown in Fig. 2. There was great difficulty in obtaining a surface free from undulations on the upper back part of the half bonnet. This part of the component is almost flat and is scarcely drawn at all; in addition, the tool did not have tension flanges. In spite of the use of a new Clearing press with excellent guides and locating points for the pressure plate upon the die (the ram itself was guided by slides within the pressure plate) it proved practically impossible to obtain constant bearing of the pressure plate upon the die throughout one series of draws.

It appeared that surface undulations could be avoided by increasing the surface of the blank, keeping the specific pressure of the pressure plate within and maintaining adequate stretching during the press operation. However, despite these changes the slightest variation in the characteristics of the sheet, or even in the distribution of the lubricant, caused a recurrence of the defect. Incidentally, this defect is difficult to correct, even by manual retouching. In addition, every time the tools were mounted in the press, a lengthy adjustment of the bearing was necessary. A better result was obtained by using cross-bars.

Surrounding the part by a flange also effected an improvement. The flange has a braking effect upon the slip and reduces the influence of surface state, whether slip or holding, in giving rise to localized stretching. Moreover, it is easy to adjust the amount of braking by cutting down the height of the flange or by increasing the entrant radii of the groove in which it lodges.

Finally, it was established that for successful production the design of the bonnet had to be altered to eliminate the flat sections at the back, slip while was regulated by a limited use of

flanges and retaining cross-bars. Tests carried out later on the bonnet of the Simca 6, see Fig. 3, showed that production in one piece was a better solution. Even if this entails an anneal, the total cost is still less than that for drawing two pieces and then assembling them by welding.



Owing to localized stretching in the rear boot, the roof panel, which was pressed from 1-2 mm sheet, gave considerable trouble. This part is shown in Fig. 4. Because of the irregularity of the results obtained at the outset, investigations were carried out to develop a metal with improved elongation characteristics. The tool, shown in Fig. 5, was completely orthodox and was mounted in an old Lake Erie press, on which the guiding was not adequate. There was no provision for regulating the pressure, 75 tons, and at the outset it was necessary to use shims for varying, although somewhat inaccurately, the pressure of the pressure plate.

For the first production batches the blank was annealed to remove the effects of cold working during the smoothing operation. In addition, there were two or three inter-stage anneals. Finally, as a result of better adjustment

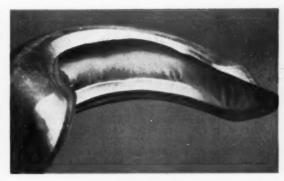


Fig. 8. Front wing drawn in one operation in Al-Cu 2-Mg

of the bearing of the pressure plate upon the die, it was possible to eliminate the preliminary anneal and all the inter-stage anneals, except for one carried out between 40 mm and 15 mm (1.5-0.5 in) from the bottom of the draw. This anneal should be carried out at a stage as far as practicable from the final draw, in order to give the finished part the maximum cold working. Fig. 6 shows the folds that form in the course of deep drawing. final anneal and the removal of the snims supporting the pressure plate allowed the folds to be taken up and the boot and the rear window to be formed.

The results obtained in the production of the roof (1.6-2.0 per cent scrap) allow the following conclusions to be drawn:—

- (a) Widening the blank, with a reduction in the relief of the flanges, a better adjustment of the bearing of the blank holder upon the die, and with more careful guiding of the tool or the press would without doubt make it possible to draw this part in one operation without an anneal.
- (b) The prospects of success would nevertheless be improved by an interruption and resumption of the

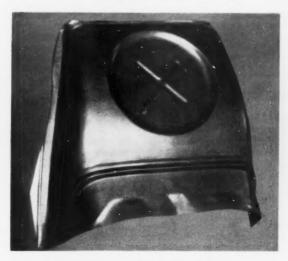


Fig. 9. The rear end of the Dyna Panhard

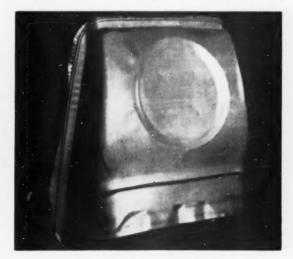


Fig. 10. A typical failure for the drawn rear end

draw at 50-60 mm (2-0-2-4 in) from the bottom. This is possible only on a hydraulic press. At a predetermined point the down stroke of the ram is halted and the pressure plate is released; the ram then descends again to complete the draw. The success of this method is probably due to relaxation of stresses within the metal.

(c) Lubrication is effected by atomization of a thin mineral oil which gives satisfaction under present conditions. Deep drawing in a single operation will probably necessitate lubrication with tallow. The economy effected through a reduction in the percentage of defects would largely compensate for the labour cost involved.

(d) At the present, the drawing of this roof in a single operation substantially represents the limit of possibilities offered by Al-Mg3.

Dyna Panhard wings

There is no difficulty in producing the rear wing, but the front wing, which is made in Al-Mg3 and is shown in Fig. 7, presents serious difficulties, as much through its shape as through its depth. Despite the following operation sequence:

(1) Draw to a depth of 65 mm (2.5 in)(2) Salt bath anneal at 400 deg C for

three minutes and cool in water
(3) Draw to full depth

a normal quality metal (base 99.5 per cent aluminium) rolled on a two-high mill gave a high proportion of parts with "bananas," that is, there were wrinkles in the part itself. A test with sheet rolled on a four-roll United mill gave better results, but there were still frequent runs of "bananas."

Finally the Al-Mg3 obtained from 99.7 per cent aluminium, rolled on the United mill, was used. With this, the concave portions of the part, where the metal did not always bear on the ram, were completely filled out. The following drawing sequences were tried and gave virtually similar results:—

(a) (1) Draw to 60 mm (2.4 in) from the full depth, with maximum pressure on blank holder

(2) Anneal

- (3) complete the draw with blank holder pressure slightly eased.
- (b) (1) Draw to 30-45 mm (1·2-1·7 in) with light pressure on blank holder
 - (2) Anneal
 - (3) Complete draw with heavy pressure on blank holder.

It was even possible to draw the blanks in a single operation, with only a few slight "bananas." Nevertheless, the difficulties of mass production and the high cost of the waste material made it preferable to have two draw stages with an inter-stage anneal. Fig. 8 shows a front wing drawn in one operation.

Dyna Panhard rear end

Al-Mg³ was used for the rear end shown in Fig. 9. Before the drawing operation the blank was folded into U shape for insertion in the blank holder. Two difficulties were experienced in drawing this part; they were:—

(1) Lorg folds were formed

(2) Splits occurred as shown in Fig. 10 and round the recess for the spare wheel.

A first pass brings the draw to the level of the reverse draw. Lubrication was by means of thin oil. After the first draw the part was torch annealed, following which a complete draw takes up the folds, while forming the housing of the spare wheel.

In spite of its apparent simplicity, it was seldom that this part could be drawn successfully either in a single operation or in two operations without an inter-stage anneal. In spite of all precautions, the part split, see Fig. 10, for stretch in the order of only 12 to 14 per cent longitudinally and 5 to 7 per cent transversely. The desired result could probably be obtained through the use of a triple-action press which would allow the spare wheel housing to be drawn at the outset and before the rest of the part.

Conclusions

The conclusions that may be drawn from practical experience in the mass production of light metal pressings for automobile bodywork are:—

(1) The tools, and more particularly the bearing surfaces between the

die and the blank holder should have a good finish.

- (2) The low elongation value of Al-Mg3 in comparison with steel must be compensated for by arranging that the metal can slip between the blank holder and die to feed any parts of the component that may be heavily stretched during the draw. The amount of excess metal left on the periphery of the blank depends on the shape of the component. It is not necessarily constant over the whole of the periphery, nor from one part to another. Splitting may occur at very different values. For example, on the rear end of the Dyna Panhard it occurs at 12 per cent elongation whereas on the bumper for the Ford Vedette it does not occur until 27 per cent elongation. This splitting technique has one disadvantage; it allows more pronounced wrinkling on the periphery with a consequent development of " bananas."
- 3) It is advantageous to provide as extensive a contact as possible between the ram and the blank during the draw. In point of fact, this tends to prevent the formation of wrinkles. It also reduces the possibility of splitting when the draw is such that the amount of stretch approaches the elongation limit.
- (4) Generally it is necessary to use cross-bars. They should have as gentle a profile as possible and seat in hollows with large radii. The height should be approximately two-thirds the height of cross-bars for the same draw in steel.
- 5) Partial torch annealing or complete furnace annealing should be carried out at such a stage as to remove the effects of cold working and so reduce the danger of folds and "bananas," and to obtain suitable mechanical properties for the part.
- (6) In many cases pre-forming of the part will eliminate the creation of folds and wrinkles during the draw. This technique is seldom used for steel parts, but it will probably be found necessary for deep drawing applications in light alloys.

THE AUTOCAR

Coronation Issue

OUR associate journal, *The Autocar*, has published a special "Coronation" and "British Cars for the World" number on the 8th May. It included features of interest to both home and overseas motorists.

In it may be found a comprehensive Guide to London's traffic and parking arrangements during Coronation week, and a summary of the regulations affecting travel by car from overseas countries to England during the period. Numerous pages have been devoted to showing the advancement of motoring from the days of Queen Victoria and King Edward VII; and a four-page feature, in full colour, illustrates the Royal residences and surrounding countryside at Balmoral, Buckingham Palace, Sandringham and Windsor.

In the "British Cars for the World" section of this special issue, overseas buyers may find detailed specifications and prices of all British cars now in production. This information has been compiled from technical data supplied

by the manufacturers themselves, and from *The Autocar's* own Road Tests. A 16-page, two-colour historical feature is also included. It gives a brief history of the British companies whose cars are now in current manufacture, with notes on the development of their models over the years.

The issue has a highly attractive, full-colour cover in gold with a simple, but effective, motif representing the journal's 58 years of service. The price remains unchanged at 1s.

FRANKFURT MOTOR SHOW

A Review of the Exhibits and German Trends in Design

By K. B. Hopfinger, M.S.A.E., M.S.I.E.

ECAUSE of general trends in Western German industry, the Western German mades, 1953 Frankfurt Motor Show was much more interesting than the only other post-war German show, which was held in 1951. In the period between the shows, Germany's trade and industrial position was considerably improved. It was in that period that the basic industries, particularly coal and steel, were reorganized to meet immediate needs and to allow the building up of moderate stocks. As a result of the improvement, manufac-turers have been able to plan for maximum output from existing plants and even for expansion of production programmes. Fiscal concessions granted by the German Government to industry for development schemes, and the availability to the German manufacturer of a certain percentage of foreign currency earned in export transactions, resulted in considerable imports of foreign-made special machine tools and equipment. Although the number of unemployed in Germany is at present well over one and a half millions, and this figure is continually being increased by refugees mainly from the agricultural parts of Soviet-controlled Eastern Germany, there is at present an acute shortage of skilled labour in Western Germany. creates certain production problems in the automobile industry, and to-day almost all firms in the German motor

and allied industries find it at present necessary to operate on a double or even treble shift system to maintain the current output, which last year totalled 428,393 cars and commercial vehicles in comparison with an output of 374,151 vehicles in 1951.

According to the production figures of the German motor industry for the first three months of this year, and the opinions of German manufacturers exhibiting at the Show, it is evident that the total output

achieved in 1952 is not likely to be appreciably increased this year, even if the export demands continue at the present level, in fact a reduced demand for vehicles is already being noted in the German Home market. This point was emphasized by Herr Max Thoennissen, the President of the German Society of Motor Manufacturers, in his speech during the opening ceremony of the Show, when he mentioned that the industry will only maintain the present output of private cars if the rate of taxation can be lowered to encourage the purchase of cars, since there is little likelihood that sales will be increased in foreign markets as long as the majority of all exports are governed by trade agreement. Herr Thoennissen also advocated the holding of only one annual International Motor Show staged each year. Successive shows would be held in different Continental capitals. Apart from questions of national prestige, this seems worthy of consideration.

This comprehensive show of private cars, buses, coaches, commercial vehicles, components and accessories, was housed in 19 large Halls, and many of the commercial vehicles were exhibited in the open. With only 43 foreign manufacturers amongst 584 exhibitors, the Show was distinctly German, although 16 different makes of American cars, 14 makes of British, five makes of French, two makes of

Italian, and two makes of Czechoslovakian cars gave the show an international flavour and it was possible to make comparisons.

Car section

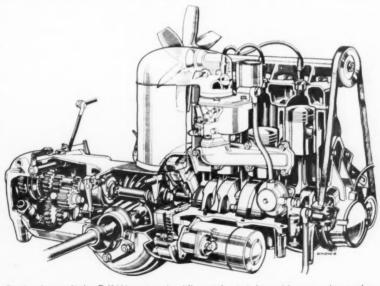
Although according to official statistics only seven per cent of all cars registered in Germany are privately owned, in this Show the emphasis was on cars with an engine capacity below two litres. This is brought about by the heavy road tax levied in Germany (which, for instance, in the case of a three litre car is approximately £49 as against a 1-2 litre car which is taxed at approximately £16), the high insurance premium due to the heavy rate of accidents, and taxation on fuel and oil which at present brings the price of a gallon of premium petrol to the equivalent of 5s. 8d., and the price of a gallon of motor oil to the equivalent of 18s. 3d.

The general trend in German car design at present is towards the small family saloon, with an engine performance allowing sustained cruising speeds in the region of 63 m.p.h. on roads like the Autobahn, and with wheel suspensions which will ensure a moderately comfortable ride on the other and generally indifferent Continental roads.

While there is definite evidence that some of the motoring public have a very keen interest in miniature cars of two

four - seater or variety, powered with engines below 600 c.c., it is interesting to note that last year's output of cars in this class amounted in Germany to only 7.392 vehicles, or 2.4 per cent of the total German car output for 1952. According to German manufacturers of minia-ture cars, they are competing mainly for customers who so far have purchased motor cycles with side cars.

Perhaps the most interesting new exhibit in the car section from the technical point of view, was the



On the three-cylinder D.K.W. engine the differential unit is housed between the gearbox and clutch



The installation of the three-cylinder engine in the D.K.W. "Sonderklasse" saloon

latest product of Auto-Union G.m.b.H., the front wheel driven D.K.W. car known as type "Sonderklasse." It has a three-cylinder in-line 896 c.c. watercooled two-stroke engine with a compression ratio of 6.5:1 and developing 34 b.h.p. at 4000 r.p.m., with a maximum torque of 51 ft-lb at 2000 r.p.m. This reversed scavenged engine (system Schnuerle) has a bore of 71 mm and a stroke of 76 mm, light alloy slightly domed-top pistons with offset gudgeon pin centres and spherical combustion chambers with a central positioned 18 mm spark plug. The engine, thermo-syphon cooled, has a light alloy cylinder head with substantial water jackets, and is held in position by 8 cylinder head bolts, equally spaced on either side of the cast iron monobloc engine block.

À noteworthy feature in the design of the engine is that the well-proportioned water chambers surrounding the cylinder bores extend uniformly over the entire length of the cylinders. The crankshaft is mounted in four roller bearings; needle roller bearings are employed for the big ends. A single plate clutch with 41 sq in frictional surface area is fitted to the flywheel which also incorporates the starter ring. The cast light alloy dry sump with specially designed shaft seals for the crankshaft, seals the crankcase which, owing to the well scavenging system employed, is pressurized. A specially designed fuel pump with a spring-loaded diaphragm

of plastic material in a die-cast body, is attached to an orifice leading into the crankcase, the pump being operated by the pulsation in the crankcase.

The engine operates on Petroil mixture in the ratio of 25 parts of petrol to one of oil and is equipped with a Solex 40 PBIC downdraught carburettor. Battery ignition is employed with an individual ignition coil for each of the three cylinders, the contact breaker being crankshaft mounted on the forward end of the engine. The six-volt generator is driven by the same V-belt as the fan pulley, the radiator being located at the rear of the engine block. The six-bladed fan is mounted on a shaft housed in a tunnel extending over the entire length of the

cylinder head. This engine, mounted forward of the front axle, is attached to the differential housing. The drive shaft is taken from the clutch through the differential housing to a locking free-wheel arrangement incorporated in the three forward and one reverse gearbox. The second and third speeds are synchronized and the pinion of the reverse gear is positioned in such a manner that the engagement of the reverse gear automatically locks the free wheel. The front wheel drive and other chassis details such as the independent front and rear suspension are identical with those of the two-stroke engine model "Meisterklasse," introduced originally

in August 1950. The only alteration to the body of the new model "Sonderklasse" is a slightly modified radiator grille, and in the case of the two-door limousine, the rear window and rear quarter-lights have been enlarged in line with the transatlantic "hard-top" style.

The Opel "Rekord"

The latest addition to the range of Opel cars, the Opel "Rekord," is the first major post-war innovation from the General Motors plant at Rüsselsheim. The mechanical details of this car are based on the Opel "Olympia," a popular model originally introduced in 1935. The wheel base of the new model has been increased by 4½ in, to 8½ in and the kerb weight reduced from 2031 lb to 1973 lb.

The production of the Opel "Olympia" is to continue for the time being. The swept volume of the fourcylinder push-rod operated o.h.v. engine has been increased from 1470 c.c. to 1488 c.c., and the compression ratio has been increased from 6-15 to 6.31; the bore and stroke of 80 mm × 74 mm remain unaltered. Power output is increased from 39 b.h.p. at 3700 r.p.m. to 40 b.h.p. at 3800 r.p.m. with a maximum torque of 70 ft-lb at 1900 r.p.m., instead of 65 ft-lb at 2000 r.p.m. A modified engine which is to be fitted to the export model of the "Rekord" has a compression ratio of 6.6:1, and an output of 51 b.h.p. at 4000 r.p.m., with a maximum torque of 78 ft-lb at 2200 r.p.m. A single plate clutch with slightly increased friction surface area is still employed, but the conventional multi-spring arrangement is now replaced by a disc type clutch spring (system Belleville).

The second and third gear of the three-speed forward and one reverse gearbox are synchronized, and a steering column gear shaft is employed. The gear ratios are, first gear 3-57:1, second gear 1-68:1, third gear 1:1, and reverse 3-57:1. There is no change in the floating hypoid bevel rear axle unit which is housed in a banjo type casing and has a ratio of 3-9:1. The new model is equipped with 13 in wheels in place of 15 in wheels as fitted to the "Olympia," the tyre size being



Pivoted rear quarter lights without pillars help to increase the field of vision from the rear of the D.K.W. "Sonderklasse" saloon

560 × 13. For the front wheels, two leading shoe brakes are now employed. Both the front and rear brakes are of the serrated cam type with automatic friction lining wear distributors. The total brake lining area on the new model is 108 sq in instead of 88 sq in.

Attention may also be drawn to the modified suspension layout of the "Rekord." For the front independent suspension, single coil springs enclosing double acting hydraulic telescopic shock absorbers are employed in conjunction with fabricated pressed steel rubber - mounted wishbone bottom links. The swan neck shaped forged stub axle has two knuckle type swivelling joints and is carried between the bottom and top wishbone links, the latter being mounted in needle roller bearings. Large size rubber bonded recoil buffers are fitted to the bottom and top wishbone links, and an anti-roll bar is fitted to the bottom links.

For the suspension of the banjo type rear axle two semi-elliptic springs with metal bonded bushes are employed. Each spring comprises three shot peened leaves and a double-acting hydraulic telescopic shock absorber of exceptional length is fitted at an acute angle from each spring towards the centre line of the vehicle, to give a certain stabilizing effect.

Although the new body is in a style that in many details has been scaled down from the current Chevrolet models, the interior layout of the two-door four-speed Opel "Rekord" falls into line with current trends in medium priced German cars. The cloth upholstery interior is rendered more decorative by chromium plated strips fitted to the interior trimming of the doors, which have no door pockets. The window area of the doors is divided by a deflecting ventilation pane, and the door window can be completely lowered by 3½ turns of the window winding handle. The rear quarter-light windows can be deflected to the outside by means of a self-locking hinged mechanism.

The instrument panel, placed directly in front of the twin spoked steering wheel, is recessed into the all-



A chromium plated strip conceals a panel joint across the rear quarter and wing of the Opel "Rekord"

metal dashboard and has a large semicircular speedometer dial centrally disposed with a smaller circular dialled cooling water thermometer, and a fuel gauge on either side. Two control light indicators for the engine oil pressure and the ignition circuit are also located on the instrument panel. Provision is made for installing a racio behind the dashboard. A large size lidded ashtray, and electric clock fitted into the glove box lid, complete the dashboard layout. The steering column gear shaft, together with the direction indicator switch, is shrouded by a metal cover. An umbrella handle type hand brake lever complete with the pull mechanism is mounted parallel to and adjoining the steering column.

Borgward exhibits

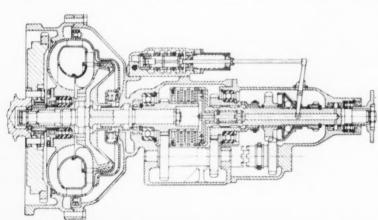
Carl F. W. Borgward G.m.b.H. of Bremen, whose entire plant was completely destroyed during the war and only rebuilt within the last four years, displayed a car developed from the recently introduced aerodynamically styled "Hansa 2400," with the rear end of the body based on pre-war research work by Prof. Kamm at the University of Stuttgart. They also showed a new four-door Pullman

limousine conversion, with a conventional style "notch back" luggage boot. In both vehicles, the all-steel metal body is of chassisless construction, the major body structure being formed by two rectangular sheet steel sections extending from just behind the radiator grille towards the rear of the body. These two rectangular sections serve also as distribution ducts for the heating and ventilating system of the venicle.

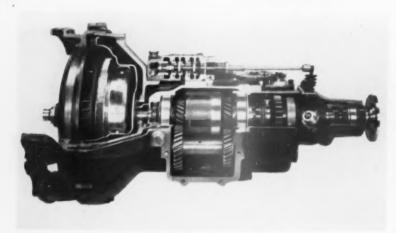
The six-cytinder push rod overhead valve engine has a swept volume of 2337 c.c., the bore and stroke are respectively 78 and 81.5 mm. With a compression ratio of 6.9:1, the engine develops 82 b.h.p. at 4000 r.p.m. and has a maximum torque of 118 ft-lb at 2400 r.p.m. The three speed forward and one reverse synchromesh gearbox has a first gear ratio of 3.01:1, second 1.59:1 and third 1:1. The reverse ratio is 3.93:1. The rear axle ratio is 3.93:1.

As an alternative to this gearbox, an automatic transmission is available for these two models, and the Borgward "Hansa 2400" is the only German made production car available with automatic transmission. At the 1951 Frankfurt Motor Show, Borgward introduced an automatic transmission for a car powered by a four-cylinder 1.7 litre engine developing 60 b.h.p. but although the performance of this vehicle was not unsatisfactory under suburban or open road traffic conditions, a certain lack of acceleration and engine power was noted in dense city traffic and on mountain roads.

Further developments within the last two years have resulted in the automatic transmission for the "Hansa 2400." This transmission is also based on a single stage hydraulic torque converter with a stalling ratio 3-8:1, coupled to a gearbox with two automatic control gears, one with a reduction of 5-1 for initial acceleration, up to approximately 28 m.p.h. When a second reduction gear with a ratio of 1-5:1 is brought into operation up to a speed of approximately 55 m.p.h., a multi-steel disc clutch arrangement



The longitudinal section of the Borgward automatic transmission



The slide valve control is mounted on top of the Borgward "Hansa 2400" automatic transmission

incorporated into the gearbox automatically engages, forming direct drive from the engine to the axle. A manually operated reverse gear is also incorporated in the gearbox.

Automatic transmission

The most interesting detail of the new Borgward automatic transmission is the hydraulic mechanical automatic control mechanism governed by a multi-stage slide valve arrangement connected by a mechanical linkage to a centrifugal governor on the output shaft. This governor comes into operation with the second gear and controls the engagement of the multi-steel disc clutch for the direct drive. The linkage connecting the centrifugal governor and the slide valve arrangement is also connected by a lost motion joint to the clutch mechanism of the direct drive. The hydraulic circuit control by the slide valve arrangement, in addition to being linked with the oil pump of the

engine, is also connected with the pump wheel circuit of the torque converter. The housing of this three-clement torque converter, the converter stator, is also the male cone of a coupling fitting into a flywheel housing of the engine. Once the second gear ratio has been reached, a second cone coupling between the converter pump and the converter turbine locks the entire torque converter, which has a limited amount of lateral sliding movement. The centrifugal governor then controls, via the slide valve at rangement, the steel disc clutch engaging the direct drive.

Another interesting feature of the "Hansa 2400" is the swing axle type suspension, with twin coil springs between the asle tubes and a pressed steel box section cross member supporting the differential housings. This differential housing has an extended pinion shaft tunnel which is attached to the chassisless body by two metal

bonded rubber cone mountings. Two similar mountings are employed to connect the box section cross member to the body.

On the Borgward stand there was also a prototype of a sports coupé powered by a 1-5 litre twin carburettor four-cylinder twin overhead camshaft engine, with a power output of 80 b.h.p. at 5500 r.p.m. The front suspension of this vehicle is conventionally arranged with wishbone links and coil springs, no anti-roll bar being employed. The rear suspension is by swing axles with deflecting trailing links and torsion bars, and is similar to the general layout employed on the Volkswagen or Porsche sports coupé.

For a more conventional four-seater sports cabriolet powered by a similar 1.5 litre, single overhead camshaft engine, Borgward also employs a swing axle but with a transverse leaf spring and radius arms. The front suspension on this car is by means of a transverse leaf spring and top wishbone links. While the Borgward "Hansa 1800" two- or four-door saloon with a four-cylinder o.h.v. engine developing 60 b.h.p. at 4200 r.p.m. has again been exhibited with only minor modifications to body details, this car is now also available with a diesel engine. The fourcylinder diesel engine with a swept volume of 1750 c.c. has a cylinder bore and stroke of 78 and 92 mm. It has push rod operated overhead valves and spherical turbulence chamber in the cylinder head in conjunction with multi-hole injector nozzles. The engine has a compression ratio of 19-8:1 and develops 42 b.h.p. at 3350 r.p.m., with maximum torque of 72 ft-lb at 1900 r.p.m.

Other car exhibits

Daimler-Benz has a production programme of ten different private cars "170 V," a car powered by a four-cylinder 1667 c.c. side valve 30 b.h.p. engine originally introduced over fifteen years ago, to the fast Mercedes-Benz "300 S." The "300 S" has a three-litre overhead camshaft sixcylinder engine developing 150 b.h.p. at 5000 r.p.m. with a maximum torque of 196 ft-lb at 3800 r.p.m. It was shown for the first time at the 1952 Paris Motor Show. While the Company did not announce any mechanical changes to the comprehensive car programme, certain minor modifications have been made to body details and the Mercedes-Benz "300 S' exhibited with a fixed head coupé attracted considerable attention. kerb weight of this vehicle, which has a 9 ft 5 in wheel base and a 19 gallon fuel tank, is 3747 lb. As a matter of comparison, it is interesting to note that the pre-war equivalent of this model, the Mercedes-Benz "540 K," had a supercharged 5.4 litre eightcylinder engine developing 180 b.h.p., but the kerb weight of the two-seater coupé was 5712 lb.

B.M.W. exhibited the latest version of the 501, a car originally exhibited



The four coned points for the rubber mounted rear suspension unit is clearly visible in this illustration of the Borgward "Hansa 2400"

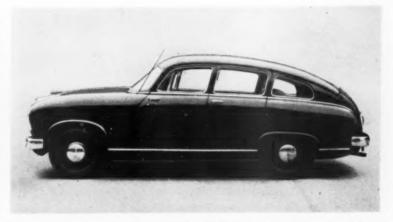
for the first time as a prototype at the 1951 Frankfurt Motor Show, but which has only recently reached the production stage. The vehicle has a six-cylinder push-rod operated engine with a swept volume of 1971 c.c., the bore and stroke being 66 x 96 mm and the compression ratio 6.8:1. Power output is 65 b.h.p. at 4400 r.p.m. and there is an almost constant torque of 95 ft-lb between 1600 and 2500 r.p.m. This engine characteristic makes it possible to obtain smoother acceleration in top gear (fourth) from less than 15 miles an hour to maximum speed, something normally not possible with a power unit of such size. An interesting design detail noted on the engine is the total enclosure of the Solex downdraught carburettor PAAI) in a large sized air cleaner and silencer. This not only ensures effective silencing of the carburettor, but also definite cooling for the carburettor by a circulating airstream. The engine is mounted apart from the gearbox in a box sectioned chassis reinforced with tubular cross members.

Hydraulic clutch control

It is of interest to note that the single plate dry clutch is hydraulically controlled. In fact this is the only German production car to be equipped with a hydraulic clutch control mechanism. The clutch control mechanism is manufactured by Alfred Teves A.G. and is combined in a unit with the master cylinder of the hydraulic brake system. The four forward speed and one reverse speed gearbox, manufac-tured by Zahnradfabrik Friedrichshafen, controlled through a steering column mounted gear shift, is fully synchronized. The gear ratios are first 4.24:1, second 2.35:1, third 1.49:1, fourth 1:1 and reverse 5.38:1. hypoid bevel differential in a banjotype rear axle, has a ratio of 4.25:1.

Front suspension is by wishbone links and double-acting telescopic dampers. The bottom wishbone links are connected to adjustable round torsion bars which are of substantial length and are positioned parallel to the side members of the chassis. The steering gear is of pinion and bevel segment type, with a reduction ratio of 16.5:1. Torsion bars are also employed in conjunction with doubleacting telescopic dampers for the rear suspension. The banjo rear axle housing is located by rubber-mounted triangular-shaped radius arms positioned at the base to the rear cross member of the chassis frame and the location points are selected in such a manner that a progressive action in the movement of the suspension obtained. The vehicle is equipped with 550×16 tyres. Two leading shoe brakes are used for the front wheels and conventional brakes for the rear. Automatic friction lining wear adjusters are fitted on all brake shoes. The brake drum diameter is 11 in and the total brake lining area 130 sq in.

The full width type of four-door allsteel saloon body has well proportioned

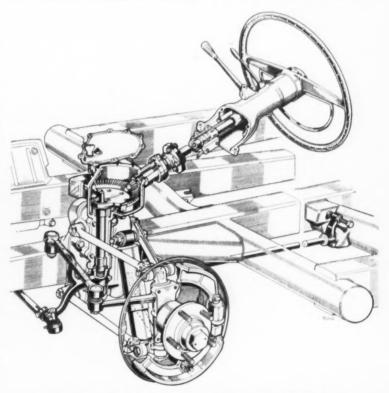


The aerodynamic shape of the Borgward "Hansa 2400" lends itself to the incorporation of a large luggage boot

window area and a fresh air heating and ventilating system developed by Gerätebau Eberspächer of Esslingen/Necker. It has an exceptionally large sized luggage boot. A 13 gallon fuel tank is located behind the rear seat squab and above the rear axle. The B.M.W. type 501 has a wheelbase of 9 ft 3 in and overall length of 15 ft 4 in, a width of 5 ft 9 in and a height of 5 ft 2 in and a kerb weight of 2832 lb.

The small front-wheel driven fourseater Lloyd car manufactured by a company belonging to the Borgward group, is powered by a two-cylinder air-cooled 386 c.c. engine developing 13 b.h.p. at 3750 r.p.m. This car, which was hitherto available only with a fabric covered plywood slab-sided body, is now equipped with hydraulic brakes. An export version with metal-covered body doors and body sides was also shown. The kerb weight of this export model is 837 lb, just over 65 lb more than the fabric-covered version. In each case the wheel base of the vehicle is 6 ft 6½ in.

Goliath, also a company belonging to the Borgward group, continue to use for their front-wheel driven car a two-cylinder water-cooled two-stroke engine with a swept volume of 688 c.c.



The front suspension and steering arrangement of the B.M.W. type "501"



At the base of each front wing fillet of the Mercedes Benz "300S" there is an intake for the ventilation system

and a compression ratio of 6-4:1. This engine has an output of 25 b.h.p. at 4000 r.p.m. and a maximum torque of 39 ft-lb. They are also offering as an alternative an engine equipped with a fuel injection system manufactured by Robert Bosch G.m.b.H. of Stuttgart. In this case, the engine has a compression ratio of 7.7:1 and the power output is increased to 29 b.h.p. at 4,000 r.p.m. with a maximum torque of 48 ft-lb. Hitherto Goliath employed a crash-type gearbox, but a new fourspeed forward and one reverse gearbox with synchronization for all forward gears is now employed. The gear ratios are for the first gear 3-28:1, second 1-86:1, third 1-12:1 and fourth 0-82:1, and reverse 3-5:1. The final drive ratio is 6-17:1. Goliath now have two leading shoe brakes for the front wheels, the brake drum diameter being 9 in and the total brake lining area has been increased to 94 sq in. The kerb weight of the vehicle is 1,980 lb.

There were remarkably few examples of special car bodies, a feature normally very much associated with Continental shows. Some coach-building firms did, however, exhibit popular German cars with special body fittings. For instance, Karossieriewerk Rometsch of West Berlin showed a normal two-door Volkswagen in a four-door version, which is gaining popularity amongst German taxi fleet operators.

Commercial vehicles

In this section almost 1,000 commercial vehicles, buses and coaches, special purpose and municipal service vehicles were exhibited. Never before at any motor show have so many commercial vehicles been exhibited and yet the only foreign exhibits in this section were those of Austin and Commer from the United Kingdom and Steyr from Austria. It would be far beyond the scope of this article to mention all the new technical details noted in this section or even list all the new models of commercial vehicles, buses and coaches exhibited. For this

reason it is proposed to deal only with some exhibits denoting definite trends of development.

Amongst the many new light commercial vehicles a 1 ton forward control van of chassisless construction was introduced by the German Ford Company of Cologne. The vehicle has a very modern appearance with a characteristic deeply curved single-piece windscreen. It has a steering column gear shift and a four-cylinder side valve 1.4 litre petrol engine developing 38 b.h.p. at 4250 r.p.m. identical to the one fitted to the German Ford "Taunus" car. As a rear loading van the vehicle has a rear bulkhead behind the cab and a body capacity of 187 ft3. The front and rear axle loads are identical and the kerb weight is 2287 lb. This vehicle was also exhibited with an all-steel truck body and another version was an eight-seater all-steel station truck with a large wellproportioned window area.

Interesting trends in small diesel engined commercial vehicles were typified by a considerable number of exhibits. Amongst these was a 1½ ton van by Ostner Fahrzeugwerke of

Salzburg, powered by a four-cylinder diesel engine manufactured by the German International Harvester Company and developing 30 b.h.p. at 2600 r.p.m.

It was noted that several other of the smaller German commercial vehicle manufacturers were employing, for light commercial vehicles, diesel engines manufactured by the German International Harvester Company of Neuss-Düsseldorf, which is also producing a 1-7 litre three-cylinder diesel engine developing 26 b.h.p. at 2200 r.p.m. This engine is also employed in certain German-made agricultural vehicles.

Borgward also exhibited various 1 ton commercial vehicles, including three different types of van, a truck and a twelve-seater coach, each powered by a four-cylinder diesel engine with a swept volume of 1758 c.c. and an output of 42 b.h.p. at 335 r.p.m.

An interesting addition to the Hanomag range of commercial vehicles is a 1½ ton diesel-engined "pick-up" truck with a steel body, the front end and cab of which have a distinctive modern appearance. This vehicle, designed specifically for the export market, has a synchromesh gearbox with a steering column gear shift, and a fresh air heating and ventilation system for the cab, which is noise and heat insulated. This truck, like all other Hanomag commercial vehicles of 1½ and 2 tons, is powered by a four-cylinder diesel engine which has a swept volume of 2799 c.c., an output of 50 b.h.p. at 2800 r.p.m. and a maximum torque of 112 ft-lb at 1800 r.p.m.

Heavier commercial vehicles

A notable feature in the commercial vehicle section was the number of vehicles up to 5 tons which since the last Frankfurt Motor Show had their carrying capacity increased. This illustrates once more the fact that not only German, but Continental vehicle operators generally, in an endeavour to obtain maximum economy, are

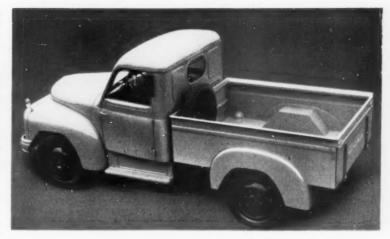


The German Ford one-ton station truck of chassisless construction

prepared to operate, particularly on short distance haulage, vehicles that will give maximum fuel economy, even though this means a certain sacrifice in the power to weight ratio.

A typical example in this category of vehicles is the new "Mercedes-Benz L 450" 4½ ton truck, which is equipped with the same six-cylinder engine developing 90 b.h.p. at 2800 r.p.m. and gearbox as the Mercedes 31 ton truck. The new vehicle has a different chassis frame and an air pressure assisted hydraulic brake system, and the rear axle ratio has been altered from 5.72:1 to 6-83:1 and the tyre size increased from 7-50 by 20 to 8-25 by 20. Another new vehicle is the "Mercedes-Benz L 5500," a $5\frac{1}{2}$ ton truck with the same six-cylinder engine, developing 120 b.h.p. at 2100 r.p.m., gearbox, front and rear axle as the "Mercedes-Benz L 5000" 5 ton vehicle; the only major difference in the "L 5500" being the revised braking system with hydrauli-cally-operated brakes for the front wheels and air-operated brakes for the rear wheels. Another addition to the extensive range of Mercedes-Benz commercial vehicles was a develop-ment of the Unimog four-wheel-drive cross-country vehicle as a track tractor with semi-trailer. A similar outfit was also shown as a municipal vehicle equipped with articulated mechanical garbage collector.

The only underfloor engined goods chassis so far produced in Germany are manufactured by Büssing of Braunschweig who, in addition to their established range of 6, 8 and 12 ton chassis. have now introduced a 4 ton vehicle powered by a six-cylinder 5-43 litre engine with a bore and stroke of 96 by 125 mm, a compression ratio of 14.5:1. The unit develops 100 b.h.p. at 2600 r.p.m. and is mounted amidship. A five speed forward and one reverse synchromesh gearbox with a steering column gear shift is employed. The chassis frame of the vehicle has parallel and almost straight side members with no projections above the top flange. The radiator and the engine air intake cleaner are located at the forward end immediately behind the first chassis Air-operated frame cross member.



The Hanomag 1½ ton pick-up truck



A new variant of the Unimog cross country chassis is adapted for garbage collection

brakes are fitted. The hand-brake pull-up type lever mounted on the chassis side member is connected to an internal drum brake located in front of the differential housing and an exhaust brake (system Haller) operated by a foot lever. A similar version of this chassis with outriggers extending from the chassis frame side members was also exhibited as a bus or coach chassis.

Noteworthy amongst goods vehicles was the range of recently introduced chassis exhibited by M.A.N. of Nürnberg, which includes a 7 ton chassis powered by a six-cylinder exhaust turbo supercharged engine with a swept volume of 8-23 litres and an output of 150 b.h.p. at 2000 r.p.m. and a maximum torque of 462 ft-lb at 1300 r.p.m.

A similar engine, but without the exhaust turbo charger, developing 130 b.h.p. at 2000 r.p.m. with a maximum torque of 376 ft-lb at 1200 r.p.m. is employed in a 5 ton chassis. By employing an exhaust turbo supercharger, M.A.N. have increased the output of their V8 diesel engine with a swept volume of 11-63 litres from 180 b.h.p. at 2000 r.p.m. to 230 b.h.p. at 2000 r.p.m., and the maximum torque has been increased from 506 ft-lb at 1200 r.p.m. to 680 ft-lb at 1050 r.p.m. Either of these engines is available for an 8 ton chassis.

The M.A.N. exhaust turbo superchargers are water-cooled, lubricated by the engine system and flanged directly to the exhaust manifold. The



A long duct connects the engine impulse with an air cleaner behind the front bumper of the Büssing 4 ton chassis



The Büssing 8 ton truck chassis

nickel chrome alloy exhaust rotor is linked by a short shaft mounted in roller bearings protected by special seals with the light alloy air rotor. Operating experience with these exhaust turbo charged engines has proved that the exhaust silencer can be simplified and reduced in size. During a trial run with the 150 b.h.p. 7 ton truck it was noted that even under full accelerating conditions the noise of the exhaust driven supercharger which revolves with the maximum speed of 40,000 r.p.m., was hardly audible above the engine noise.

The only other exhaust turbo supercharger at the Frankfurt Show was exhibited by Gerätebau Eberspächer of Esslingen/Necker. This exhaust turbo charger is designed to be fitted to existing engines with an unsupercharged output of 70 to 150 b.h.p. and a maximum exhaust temperature of 1256 deg F. The rotor design of this unit is such that an increased power output of up to 25 per cent can be obtained from an engine without any alterations to the valve overlap. This exhaust turbo charger is self-contained,

has a maximum operating speed of 3 8 0 0 r.p.m., a n overall diameter of 11 in and an overall length of 9\frac{1}{4} in. It is splash lubricated and aircooled and weighs a p proximately 37 lb.

Because of the increasing demand for four - wheel drive vehicles by countries participating in the European defence programme, and also by constructional operators on the Continent and abroad, most of the major German commercial vehicle manufacturers are now producing such vehicles. Magirus Deutz,

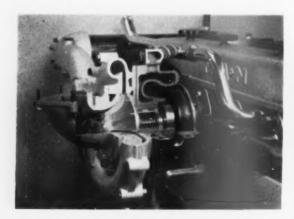
vehicle has a turning radius of 26 ft 4 in. This truck is equipped with an air-cooled V8 cylinder 10-6 litre diesel engine developing 175 b.h.p. at 2250 r.p.m., with a maximum torque of 433 ft-lb at 1500 r.p.m. The weight of the power unit complete with single plate clutch but less the six-speed Z.F. gearbox is 1873 lb.

Henschel also exhibited a four-wheel drive 4-5 ton truck powered by a six-cylinder 5-42 litre 100 b.h.p. oil engine and also a 5 ton four-wheel drive truck with a six-cylinder oil engine having a swept volume of 6-16 litres and an output of 115 b.h.p. at 2240 r.p.m.; this vehicle had a fully fronted cab.

Kaelble exhibited for the first time an 18 ton all wheel driven dumper truck with twin rear axles. This vehicle has a V8 cylinder 19·10 litre oil engine developing 200 b.h.p. at 1400 r.p.m., with a maximum torque of 728 ft-lb at 950 r.p.m.



The installation of the exhaust turbo charger on the 175 b.h.p. six-cylinder M.A.N. direct injection diesel engine



The water cooled exhaust turbo charger of the M.A.N. 175 b.h.p._six-cylinder diesel engine

who originally produced a 3-5 ton all wheel driven truck powered by an air-cooled four-cylinder dieselengine, with a swept volume of 5-32 litre and an output of 90 b.h.p., have now also introduced a 6-5 ton four - wheel driven truck with a wheelbase of 14 ft 5 in, employing 11-00-20 twin rear and single front wheels. The

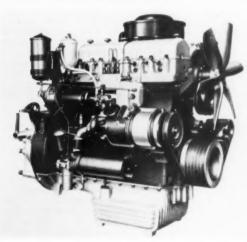
Suedwerke exhibited an 18 ton dumper truck powered by an eight-cylinder supercharged direct injection two-stroke oil engine, developing 210 b.h.p. at 1700 r.p.m. with a maximum torque of 723 ft-lb at 1200 r.p.m. The weight of this power unit complete with clutch but less the 6 ft gearbox is 2364 lb. The hydraulic tipping gear employed in the dumper trucks is produced by Meiller of Munich and has a maximum tipping angle of 70 deg, which under full load conditions can be reached within 18 secs.

Throughout the commercial vehicle section a distinct trend towards fully fronted cabs could be noted. To give increased driving comfort, many German manufacturers of heavy

commercial vehicles are now employing steering column gear shifts and are equipping the cabs with fresh air heating and ventilating systems and comfortable adjustable seats. Some cabs even have arm rests. On long distance vehicles, the general tendency in Germany and on the Continent is to use substantially larger cabs with sleeping accommodation for the crew in the form of fixed or collapsible bunks, including clothes lockers and in some cases cooking facilities, all located beyond the driver's seat.

Buses and coaches

The conclusion drawn after examining the large number of buses and coaches exhibited at Frankfurt is that German and Continental operators are very conscious of the advantages to be obtained from lighter vehicles of integral construction, but the major demand is still for vehicles with conventional chassis and bodies. Never-



The Sudwerke four-cylinder 145 b.h.p. two-stroke direct injection diesel engine

theless, it can be said that the trend in design of German buses and coaches of integral construction has advanced considerably since 1951. Generally integral construction was based on sub-structures and bodyframework, and the stressed skin method of construction was used on very few vehicles. There is a marked tendency amongst German manufacof integrally - constructed turers passenger service vehicles to employ independent wheel suspension with wishbone links and coil springs in conjunction with anti-roll devices and comparatively large size hydraulic telescopic shock absorbers.

The general German trend in bus body design is to use power operated divided folding doors, and in the larger single deck vehicles, a door also gives access to the rear overhang, serving mainly as a standing platform. In vehicles of this type the conductor's desk usually with raised seat enabling the observation of the entrance door, is arranged part way down the body



Magirus Deutz 6½ ton four-wheel drive tipper truck powered by a 175 b.h.p. V8 air cooled engine

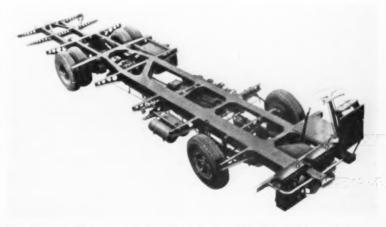
centre near the door which serves as an exit. It must be realized that this arrangement was brought about by the fact that many municipalities whose tram system was destroyed during the war, are now employing single deck buses with a carrying capacity of 100 passengers but with seating accommodation for only 30 or 40.

Double deck buses are in operation only in Western Berlin, low arches spanning streets in towns and other structural obstruc-

tions are at present severely restricting the use of this type of passenger vehicle in Germany. While German road regulations specify a maximum axle weight of 8 tons, a special dispensation has been granted to operators of 100 passenger single deck buses, which have in some instances rear axle loads above 10½ tons.

To give good all-round vision, the general trend in German coach body design is to employ large side windows and glazed quarter lights which are usually slightly tinted. There was the sharp decline in the number of coaches with folding and sliding roofs. In their place, fresh air heating and ventilating systems were fitted. In the majority of touring coaches the heating and ventilation systems are served by a fuel fired combustion heater which in some instances operates through water filled heat exchangers that are connected with the engine cooling system to permit pre-heating of the engine and body before the vehicle is put into service during severe weather.

While German opinions on the most suitable engine position vary considerably, there is now a definite trend towards the use of underfloor engines.



In the Henschel 100-passenger single deck chassis, the right-hand frame side member serves as an air duct for the heating and ventilation system



The engine is mounted transversely at the front of the Henschel bus chassis type "HS.100N"

Rear engined vehicles are exceedingly popular at present but for smaller vehicles with a passenger capacity of less than 30, the conventional forward mounted engine is still predominant.

One of the most interesting new bus chassis exhibited was the 100 passenger single deck underfloor engined chassis by Henschel and Söhne G.m.b.H., a company which in the post-war period has pioneered the transverse vertical forward engine position. The new Henschel underfloor engine bus chassis has a wheelbase of 19 ft 23 in and an overall length of 38 ft 101 in. It is powered by a six-cylinder 13-31 threelitre diesel engine employing the Lanova pre-chamber injection system and developing 200 b.h.p. at 2000 r.p.m. with a maximum torque of 194 ft-lb at 120 r.p.m. The chassis exhibited had a Wilson four-speed air-operated epi-cyclic gearbox mounted separately from the engine, to which are attached the

hydraulic clutch and overdrive gearbox. The chassis frame comprises two welded box section members, one of which is 12 in wide and 8½ in deep, serving as a backbone section, while the other member passing over the engine has a section of 5 in × 4 in. Substantial cross members are 'used, but the outriggers and the rear platform support are of very much lighter construction. The large box section side members serve also as a distribution duct for a heating and ventilating system. The total weight of the complete chassis is just under 6 tons.

Büssing displayed a new lightweight bus chassis design for integral body construction powered with a 5-4 litre 100 b.h.p. 6-cylinder underfloor engine. It has a total weight of just under 3 tons. Another interesting detail on Büssing passenger service vehicles was an underfloor engined chassis powered by a 150 b.h.p. engine equipped with automatic "Diwabus" transmission manufactured by J. M. Voith G.m.b.H. This transmission incorporates a water cooled hydraulic torque converter in conjunction with differential planetary gearing and free-wheel, and a special brake arrangement. The transmission on the vehicle exhibited weighed 537 lb. A similar Diwabus automatic transmission was also shown in a bus chassis produced by Daimler-Benz, M.A.N. and Magirus Deutz (a full description of this transmission will be published in a later issue).

Magirus Deutz exhibited for the first time a rear engined bus for a coach chassis, with a 13 ft 10 in wheelbase, powered by a six-cylinder air-cooled diesel engine with a swept volume of 7.98 litres developing 130 b.h.p. at 2250 r.p.m. The weight of the complete chassis with an overall length of 29 ft 3 in is just over 3 tons, and the total laden weight of the vehicle is 81 tons.

A 46-seater coach of integral design, with unladen weight of 7 tons and a maximum operating weight of 13½ tons, was exhibited for the first time by Firma Theodor Klatte of Bremen. The vehicle, powered by a rear mounted 10-64 litre Deutz air-cooled V8 diesel, developing 175 b.h.p. at 2250 r.p.m. is equipped with an electrically operated Z.F. "Media" gearbox and has an independent front wheel suspension

springs above and a stabilizer bar.

A hydraulic jacking system operated from an engine driven pump unit and a fuel oil burning combustion heater, both developed by the company, are incorporated in the vehicle. The same company has also been testing with considerable success for the past year a similar coach equipped with hydraulic transmission driving all four wheels.

employing wishbone links with coil

WEAR INVESTIGATIONS

N 3W techniques using radioactive tracers for investigating friction and wear are described in an article by L. T. Burwell and C. D. Strang, in Product Engineering Annual Handbook, 1953. Friction between two sliding metal surfaces is largely due to the shearing of minute local pressure welds, which results in a transfer of metal from one surface to the other. This theory has been confirmed by experiments in which the radioactive tracer method was used.

Two cylindrical specimens were pressed together with their axes at right angles to make contact at one point. One specimen, made radioactive by slow neutrons in a uranium pile, was held stationary, while the inert cylinder rotated about its own axis. The friction track thus formed was investigated with a Geiger counter which gave a quantitative measure of active metal

transferred, and with an X-ray film to obtain an indication of the distribution of transferred material.

The track obtained with the two steel cylinders consisted of discrete spots corresponding to individual welds. By lubricating the contacting surfaces of the cylinders, the transfer was reduced. This effect was found to be caused by a reduction in the size of each weld, the number of welds remaining unchanged. Transfer weight measurements indicate that the tracer technique may provide a rapid means of evaluating the galling resistance of materials.

A series of nitralloy rotating inert specimens, each of different hardness, were used to study the effect of hardness on micro-welding. It was found that the curve of transfer weight plotted against hardness rises steadily, and shows no discontinuity where the rotating member becomes harder than

the stationary one, indicating that transfer occurs in both directions. When widely different metals are used for the moving surface, the effect of composition is much greater than that of hardness. Carburized steels, although harder than the untreated ones, show less transfer because of a lower tendency of carbides to weld.

The radioactive tracer technique lends itself to the study of practical mechanical operations, such as the action of piston rings in a cylinder, or the building up of an edge on metal cutting tools. When applied to wear measurements on piston rings, for example, the method consists in analyzing samples of crankcase oil for traces of radioactive material from the rings, and has the advantages of rapidity and of showing ring and cylinder wear independently of that of other engine parts. (M.I.R.A. Abstract 6180.)

FUEL INJECTION NOZZLES

An Investigation into Causes of Corrosion

By W. P. Mansfield B.Sc. (Eng.)

THE British Internal Combustion Research Association recently received a request for assistance from a member company confronted with a serious problem in connection with engines of a modified design. A considerable number of these engines had been sold, mainly to overseas customers. After they had been in service a short time, reports were received of rapid nozzle failure due to loss of metal from the wall of the nozzle body, resulting in many cases in discharge of fuel through the nozzle wall. The problem had not been encountered in prototypes used since 1947 for the development of the engine. Most of the affected engines were operating on A Class fuels. It was later learned that the same trouble had been experienced in varying degrees of severity in many engines of other makes. In the light of some preliminary fundamental work which had recently been carried out for another purpose, the Laboratory was able to suggest the probable cause of the trouble. Experiments proved the theory correct, and showed that the trouble can be remedied easily.

Nature of the problem

The type of nozzle used in the engine in question was the widely-used S-size long-stem nozzle. A standard solid copper washer was fitted at the cap-nut seating face. The design of the injector housing in the cylinder head was such as to provide ample cooling of the injector. Some of the returned nozzles are shown in Fig. 1.

Nozzle A, returned from Portugal, failed three months after installation of the engine. When the nozzle was tested it was found that the fuel dis-

charged through the side wall instead of the normal orifice.

Nozzle B, returned from Syria, also discharged through the side. It had been welded in an attempt to effect a repair, as this was sometimes found to be sufficiently successful to enable the engine to be kept running. Later, brazing was found to give better results, but even then success was achieved in very few cases.

Nozzle C, another of a number returned from Syria, was the only example seen by the engine makers in which the attack had extended beyond the shoulder of the nozzle to the large diameter portion housed inside the

The number of hours service completed by the above nozzles was not known, but in general failure had been found to occur in about 300 hr and sometimes less.

Formulation of the low temperature theory

With a view to making a more fundamental approach to the question of the formation of corrosive substances in internal combustion engines, which was considered to be important in connection with work on lower grade fuels, the Laboratory had recently studied research carried out elsewhere on corrosion by combustion products. The work carried out by the British Coal Utilisation Research Association on the problem of corrosion of the external heating surface of water-tube boiler plant was found of particular interest. A major fact emerging from this work was that the presence of traces of sulphur tri-oxide in the flue gas could be responsible both for the formation of bonded deposits and also

for the corrosion and blocking of the air heater and other cooler sections of the plant. Other investigators had found that the presence of a few parts per million of sulphur tri-oxide in boiler flue gas elevated the dewpoint considerably above what might be expected in relation to the water vapour content, and that the condensate at these higher temperatures was strong sulphuric acid capable of ready attack on iron and steel surfaces. B.C.U.R.A. had shown further that carbon particles absorbing sulphur tri-oxide can corrode steel surfaces at temperatures as much as 104 deg F (40 deg C) above the acid dewpoint. With the assistance of B.C.U.R.A.

With the assistance of B.C.U.R.A. some preliminary tests had been made in the Laboratory to assess the sulphur tri-oxide content of engine exhaust gases using the B.C.U.R.A. dewpoint meter. This instrument consists of a glass tube, the closed end of which is exposed to the gases under examination and progressively cooled by an internal air stream until condensation occurs at the exposed surface, this point being indicated by the passage of current between two electrodes embedded in the glass surface.

the glass surface.

All this work emphasized the fact that the temperature of the surface exposed to combustion products is a factor of first importance in this type of corrosion problem. Consequently, when two examples of the affected fuel injection nozzles were received at the Laboratory, attention was given to the possibility of the surface being overcooled. The surfaces of the nozzles had many interesting features, as can be seen from the illustrations, but the most significant was judged to be the complete absence, in both cases, of



Fig. 1. Examples of nozzles corroded in service



Fig. 2. A nozzle showing artificially induced corrosion

attack at the discharge end of the nozzles. This end being subjected directly to the heat produced in the combustion chamber, is at a high temperature while the end adjacent to the cap-nut is shielded from the heat and cooled via a short heat-flow path. It was considered very probable that the attack was due to the temperature of the nozzle surface remote from the combustion chamber being too low and it was therefore suspected that the nozzles were being cooled more effectively than usual.

Brief consideration of various theories

At this juncture the Laboratory was asked to send a representative to a meeting, arranged by the member company, with the fuel injection equipment makers. At this meeting the nozzles shown in Fig. 1 were studied together with other exhibits including carbonaceous deposits removed from various nozzles. Representatives of the injection equipm at manufacturer stated that the trot ale

had been experienced in varying degrees by many engine makers, but had not previously reached quite such serious proportions as in the present case. Various possible explanations of the loss of metal from the nozzles were considered.

Variation in metallurgical structure

Since the trouble had not occurred in the experimental engines on which the development work had been conducted, the engine makers thought that the fault lay with the material or heat treatment of the nozzles. It had been found that some

of the nozzle bodies when received from the makers could be marked with a file, others could not. The nozzle makers stated later that the nozzle bodies were of high nickel - chrome steel which could be more easily filed than carbon

steel of the same hardness number. Three nozzles had been returned earlier by the member company for examination, two had been attacked heavily in 300 and 500 hr while the third was quite unaffected after

6.000 hr service. All three were found to have a satisfactory case structure with hardness between 655 and 678 V.P.N. at 10 Kg which was within accepted limits, and there was no significant metallurgical difference between them. The hard-ness of some other between affected nozzles ranged between 634 and 707 V.P.N. The depth of the case-hardening was correct, but the casing of some of the nozzles was slightly deficient in carbon.

Electrolysis

The fact that the attack was heaviest on the surface adjacent to the copper washer naturally suggested electrolytic attack, the copper forming the anode and the steel of the nozzle the cathode. If this were in fact occurring, the substitution of a steel or an aluminium washer would effect a cure. It was learned that tests had been made with both materials but neither had made any improvement.

Gas erosion

It was believed in some quarters that the loss of metal was due to erosion by the cylinder gases moving at high velocity over the surface. It was suggested that the swirl imparted to the air during the induction stroke was greatly intensified in that part of the gases which was compressed into the clearance space round the nozzle, and that these gases scoured the surface of the nozzle. An allied theory was that the surface was abraded by particles of hard carbon carried by gases moving in to and out of the space at high

Running condition	Water outlet temperature		Washer temperature	
	deg F	deg C	deg F	deg C
Full load (for several hours)	140	60	216	102
Idling	113	45	135	57
Full load (for 15 min after idling)	140	60	198	92

lowest at the cap-nut end where the attack was greatest. Moreover, it had been reported that the clearance space near the washer where the attack occurred was invariably found filled with soft damp carbon.

A nozzle received at the Laboratory for examination some years ago had channels across the edge of the end surface facing the combustion chamber, but was unaffected elsewhere. The nozzle, which was used in a two-stroke

velocity. Against these theories it was

pointed out that the gas velocity would

be highest at the discharge end of the

nozzle where no attack occurred and

out was unaffected elsewhere. The nozzle, which was used in a two-stroke cycle engine, operated at a high temperature. It was concluded in this case that the nozzle had been eroded by gases compressed into the clearance space round the nozzle, and chamfering of the combustion chamber end of the bore in the cylinder head to reduce the gas velocity was recommended. The position of the attack and the appearance of the affected surface were entirely unlike those observed in the present investigation.

A representative of the injection equipment manufacturer recalled his first acquaintance with the effect under study. The nozzle of a swirl chamber engine was heavily attacked near the washer. When the injector was removed, a layer of carbon which had formed across the face of the nozzle remained in position at the end of the bore in the cylinder head, there being only a small hole in the middle through which the spray had passed. With this carbon layer in position there could be



Fig. 3. A nozzle with artificially induced corrosion, before and after cleaning

no question of high velocity gas movement in the clearance space round the nozzle

Leakage of nozzle washer

Two complete injectors, included in the exhibits, had a thick coating of tacky lacquer over the greater part of their external surface. This showed that combustion gases had been leaking past the nozzle washer and it had been suggested that the nozzle trouble might be associated with this condition. To disprove this, the engine maker had made a test in which every precaution had been taken to ensure gas tight joints at the washer faces, and no trace of leakage occurred. The nozzle was attacked as before. A second test had then been made using an old iron washer which leaked so much that strong puffs of gas could be felt at the end of the injector bore. No attack occurred. It was pointed out that this evidence supported the low temperature theory since the effect of the blowing of cylinder gases past the clearance round the nozzle would be to raise the temperature of the nozzle.

Low nozzle temperature

Attention was then turned to the theory that the trouble occurred when the temperature of the nozzle surface was low enough to permit chemical attack by sulphur compounds produced by combustion. It was seen that all the affected nozzles exhibited were practically free from attack at the discharge end, and that the loss of metal was greatest at the cool end. Thermocouple measurements made at the Laboratory in the copper nozzlewasher in a vertical engine otherwise similar to the horizontal engine in question were quoted. The results are shown in the accompanying table.

The fuel injection equipment makers considered these figures very low. The engine maker said the nozzle temperature in the horizontal engines in service would be even lower, and he described the cooling system of these engines. Water, from a large tank, which took some hours to warm up, entered the lower side of the cylinder block and passed almost directly to the lower side of the cylinder head where the injector was situated. The injector was quite

cold to the touch.

To test the low temperature theory further, the various cases which had come to the fuel injection equipment makers' attention were then reviewed and the cooling system considered in each case. The great majority of these cases had occurred in coastal districts and most were with marine engines. Hence it had been thought that a saltladen atmosphere might be the cause.

The many marine engines concerned were sea-water cooled directly, and hence had low nozzle temperatures. The other engines employed in coastal districts which had given the trouble were vehicle engines. In these engines the cooling water was passed to the cylinder head first, where a full flow was maintained, while only a part of AUTOMOBILE ENGINEER

the water passed, under the control thermostat, of a through the cylinder jackets. In certain other vehicle engines in which the trouble was almost unknown, though the nozzles used were taken from the same batches as those supplied to the member company, the water flow through the cylinder head was under the control of a thermostat which kept the temperature high.

In the case of the swirl chamber

engine mentioned earlier in connection with the gas erosion theory, the shortstem nozzles used were situated in a passage well back from the combustion chamber, and were cooled by a jet of cool water arranged to impinge on the injector housing in the cylinder head.

Another instance in which the trouble had occurred was in engines used in barges on the Venetian canals. Here again the cold water was probably passed directly through the cooling passages of the engines. The low temperature theory was supported by every case that could be recalled.

Since this meeting another interesting case has come to light. A member company found the same action occurring on nozzles fitted in one of the company's engines installed in the works power house. The cause was not discovered. The nozzles were replaced by new ones, and later the engine was removed from service for other reasons. In this case, normal jacket water temperatures were used but an unusual feature of the engine was an aluminium-alloy cylinder head which was being used experimentally. There seems little doubt that the high thermal conductivity of this material resulted in nozzle temperatures lower than normal.

Experimental investigation

Although the evidence in favour of the low temperature theory was very strong, it was considered desirable to obtain definite proof that temperature is the controlling factor. To this end an engine was chosen in which longstem nozzles were used but which was entirely free from the trouble, and steps were taken to produce the nozzle attack by simply lowering the nozzle temperature. In place of the normal washer, two copper washers of smaller outside diameter were fitted in front of the cap-nut leaving an annual space of section measuring 3 mm axially by 5 mm radially below the cap-nut. cylinder head was drilled to take two copper pipes through which cold water was passed at high velocity round the annular space, thus cooling the nozzle cap-nut and washer and hence the



Fig. 4. The cap nut used with the nozzles shown in Figs. 2 and 3

nozzle. In addition, cold water from the mains supply was passed through the cylinder head. The engine was run for 9 hr 20 min on Pool Gas oil at two-thirds full load and the nozzle then removed for examination. Fig 2 shows its appearance. The same type of etching observed in all the affected nozzles was present and the characteristic irregular grooves had started to form even in this short time. As with the service specimens, the end of the nozzle adjacent to the combustion chamber was free from attack.

A further test was then made with another nozzle which had previously been used for over 500 hr in the same engine without any effect on its surface. Examination after 9 hr 20 min showed that the attack was occurring but at a somewhat slower rate than in the previous case. Running was, therefore, continued to a total of 20 hr 30 min. Fig. 3 shows its appearance on removal and after cleaning. The characteristic features were again present. grooving had reached a depth of 0.36 mm, i.e. 18 per cent of the nozzle wall thickness, so that complete penetration of the wall could be expected in a total of 114 hrs running. nozzle cap-nut which had been used in both tests was also heavily attacked as shown in Fig. 4.

The nozzles used in these tests were made by a different manufacturer from those which had suffered in service. This, together with the corrosion of the cap-nut, is further evidence that the trouble is not connected with nozzle material or heat treatment.

This brief investigation sufficed to show that this type of nozzle corrosion can be eliminated simply by a suitable increase in nozzle temperature. Raising the cooling water temperature is normally an effective method of achiev-Where the use of cold water ing this. cannot be avoided, the cylinder head should be designed to ensure that a moderate nozzle temperature is attained shortly after starting the engine, On the other hand, it is well known that excessively high temperatures lead to other nozzle troubles.

EXHAUST BRAKES

A Review of Applications for Diesel Engined Vehicles

By J. L. Koffman, Dipl.Ing., M.I.Loco.E.

S a result of the continual improvement in the performance of heavy commercial vehicles, the problem of effective braking is one of increasing importance, particularly for operations in those countries where long down gradients have frequently to be negotiated. It is in such conditions that an additional brake, which would relieve the normal brakes and thus keep them cool and fully effective for emergencies, will be most appreciated, and even more so when elimination of continual operation of conventional wheel brakes reduces the strain on the

Although the engine exhaust brake satisfies these requirements and is widely used on the Continent, its use

in this country is generally not considered necessary since long gradients are few and far between on the roads normally used by heavy road vehicles. Nevertheless, tests have recently been carried out by the Fighting Vehicles Research and Development Establishment of the Ministry of Supply to determine the effect of an exhaust brake on the availability of the wheel brakes for emergencies, and on driver fatigue These tests were and brake wear. carried out on 10-ton six-wheeled lorries powered either by diesel or

Power available for braking

Essentially, the exhaust brake actuating gear consists of a valve that is placed as close as possible to the exhaust manifold to reduce the number of components subjected to the resultant pressures. The operating

cycle of the engine when the exhaust brake is in use is shown in Fig. 1. Air is aspirated and compressed in the usual manner. On the downward stroke the air is expanded and a small of amount energy absorbed by the cooling system, since the compressed air is cooled during expansion as indicated by the area A. At this point the exhaust valve opens and the air is forced into the manifold space limited by the closed brake valve.

The pressure in the manifold space is increased by

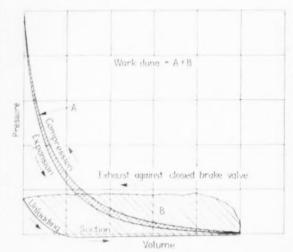


Fig. 1. Exhaust brake cycle diagram

the air forced in until it exceeds the pressure maintained by the exhaust valve spring. As the piston moves downwards during the intake stroke, the air pressure in the space between the exhaust valve and the brake valve will lift the former off its seat, and air will escape to atmosphere via the open inlet valve until the pressure behind the exhaust valve is reduced sufficiently to allow the spring to force the exhaust valve down on to its seat. done during this part of the cycle is indicated by the area B. Fig. 1 demonstrates two important points. the amount of work absorbed by the engine as a compressor, is to a certain degree determined by the setting of the exhaust valve spring and the valve timing (overlap). Second, the emission of compressed air through the inlet calls for the use of air cleaners whichif of the oil-bath type-will not spill

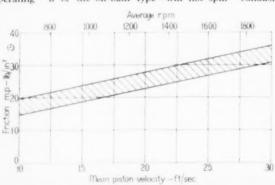


Fig. 2. Frictional losses of bus and lorry diesel engines

oil out through the inlet aperture.

The power made available by an engine working as shown in Fig. 1 depends upon the frictional losses and the work done at A and B. Values of frictional losses have been determined by a number of investigators, those for typical bus and lorry diesels are shown in Fig. 2 (Ref. 1). The power available for braking without the use of a brake valve can also be obtained as shown in Figs. 3 and 4. Fig. 3 relates to the four-stroke, six-cylinder 9-6 litre A.E.C. engine (Ref. 1) and Fig. 4 to the two-stroke, four-cylinder 5.88 litre Krauss - Maffei engine (Ref. 2). For these diagrams the fuel consumption is plotted per h.p. hour as a function of b.m.e.p.,

and fuel consumption per litre and revolution for a number of speeds. Frictional and other engine losses are obtained by extrapolation of the curves

to zero fuel consumption. At maximum speeds the friction losses of a motored engine are of the same order as those of an engine working normally, but they are appreciably lower at about half maximum speed because of the lower pressures. Pumping losses amount to about 1.5 friction losses (Ref. 3), and the total losses are approximately equal to the negative b.m.e.p. (friction mean pressure, or f.m.p.) values shown in Figs. 3 and 4. The ratios of f.m.p. to b.m.e.p. for two engines are plotted in Fig. together with the ratio for one of the engines with the brake valve shut and a maximum pressure of 35 lb/in2 maintained in the space between the

exhaust valve and the brake valve.

Values of this ratio as high as unity were recorded (Ref. 4) for a 4.56 litre, fourcylinder engine developing 52 h.p. at 2,000 r.p.m., but these values appear to be on the high side for modern engines. Generally it may be assumed that with the exhaust valve brake shut, the f.m.p. will be about twice that with the valve open, Fig. 6. The power thus made available for braking is augmented by the tractive resistance, transmission losses and the power absorbed by the

auxiliaries.

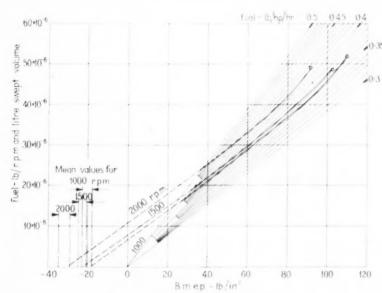


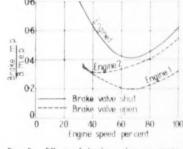
Fig. 3. Friction losses of a six-cylinder, four-stroke engine

Brake valves

The simplest form of brake valve is the butterfly unit as made by Saurer Co., of Arbon, and Haller of Zurich. Saurer use a cast housing and valve which is secured by pins on a spindle carried in carbon bearings; asbestos packing is provided at the operating end of the spindle. This type of valve is completely satisfactory with diesel engines, but with petrol engines running on leaded petrol, a lead deposit is formed inside the housing and the leading edge of the valve, which must be cleaned about every 300 miles to ensure full closure when the brake is applied.

A sliding valve developed by Oetiker of Zurich originally had two valves, a main valve and a small unloading valve. The valves were connected by links so that when closing, the main valve closed first, while the unloading valve moved first when opening, thus relieving the main valve of the gas pressure. This valve proved to be generally satisfactory for both diesel and petrol engines, but the unloading valve did occasionally stick owing to a build-up of deposit. In consequence it has been eliminated from the latest design.

To prevent any possibility of incorrect operation the valves should be



interlocked with the fuel supply. This

can be done in a simple manner as shown in Fig. 7. The brake valve lever is mounted horizontally on the dash-

board below the steering wheel. It is about 10 in long and can be moved through 90 deg. As soon as the lever

is moved through about 15 deg, the fuel supply is cut off and the springs compressed. Further movement of the

lever operates only the brake valve. If the fuel pedal is then depressed,

Fig. 5. Effect of brake valve on ratio : power to brake/power to drive

only the spring is released without

affecting the fuel pump. Return of the brake valve into the open position automatically restores the fuel supply, provided the position of the fuel control should demand this. This means that the exhaust brake must be released shortly before the vehicle is stopped if the engine is to continue running.

Operating experience

Before the tests were carried out in this country, the experiences of Swiss operators were considered in some detail. Major bus operators, such as the Postal Authority, the Chemins de Fer Fribourgeois and Berne City

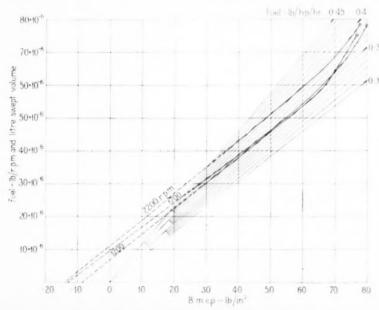


Fig. 4. Friction losses of a four-cylinder, two-stroke diesel

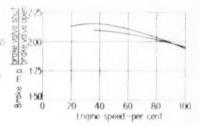


Fig. 6. Effect of brake valve on power to brake

Transport Department, were unanimous in their praise of this simple device, which was used effectively not only over the mountainous routes of the first two undertakings but also in the relatively level streets of Berne. The wear of brake shoes and drums was stated to have been reduced in most cases to about one-third of that experienced in operation without an exhaust valve.

Of the tests carried out in this country, only those obtained on diesel engined vehicles will be dealt with in these notes. The vehicle had a gross

weight of 20 tons, and was powered by a six-cylinder engine developing 120 h.p. at 1800 r.p.m. A lever of the hand brake type, placed at the driver's right, was used to operate the brake valve. It had a trigger handle arranged to cut off the fuel supply before the lever could be moved. Originally, some difficulty was experienced in sealing the manifold expansion joints. It was overcome by brazing them up.

Most of the testing was carried out over a hilly road circuit in South Wales. The circuit was about 15 miles long. It included three major gradients, one of 1 in 10.5, one of 1 in 10 and one of 1 in 8.5. The 1 in 10.5 gradient was 3-10 miles long and the other two were each 0.90 miles long, therefore the brakes were applied for about five miles in each lap, which was covered in about one-and-a-half hours. A total of 1500 miles was run with the normal brakes applied as little as possible and another 1500 miles without the use of the exhaust brake. New brake shoes were used for each test. Wear determinations were made after 500 miles and after 1500 miles.

In the tests when the exhaust brake was used, the average lining wear was 0.007 in at 500 miles and 0.011 in at 1500 miles, with corresponding drum wear of 0.006 in and 0.008 in. comparative figures for the tests in which only the wheel brakes were used were 0.013 in and 0.026 in lining wear and 0.005 in and 0.009 in drum wear. The rise in brake drum temperature was checked at the end of a 121 minute descent. It was only 20 deg F when the exhaust brake was used against 112 deg F for normal braking Throughout the tests the pressure at the brake valve was maintained at about 38 lb/in2.

Not only was there a reduction in lining wear and in drum temperature rise when the exhaust brake was used, but it was also found possible to drive the vehicle downhill faster and more confidently. In descending a steep hill with only the exhaust brake in operation, it was necessary to use the same gear as for ascending, but by using the

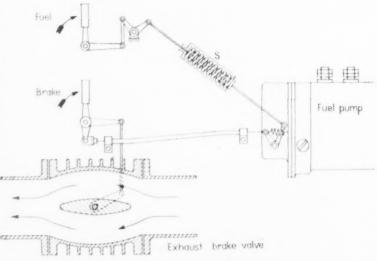


Fig. 7. Fuel-brake valve interlock arrangement

exhaust brake and making occasional applications of the normal brakes, it was found possible to use one gear higher for the descent than could be used for the ascent. This permitted a faster descent, and at the same time assured full availability of the standard brakes for emergency stopping. It was also found that the use of an exhaust brake appreciably lessened driving in the negotiation of long descending gradients. From the above data it will be appreciated that the exhaust brake is a very desirable feature for diesel engined vehicles that are to operate over hilly routes.

The author is grateful to Mr. A. E. Masters, C.B.E., Chief Engineer, F.V.R.D.E., for the encouragement given in connection with the testing of the vehicles; he also wishes to record his indebtedness to Mr. V. Diem, Director of the Saurer Co., Arbon, for the facilities placed at his disposal for the study of exhaust brake operation, and to Mr. H. Burkhardt, Chief Engineer of Swiss Postal Bus Services, Mr. Xavier Remy, General Manager

of Chemins de Fer Fribourgeois, and Mr. Thoenen of Berne City Transport for much valuable information and demonstration of vehicles in service. Acknowledgment is made to the Chief Scientist, Ministry of Supply, for permission to publish this paper.

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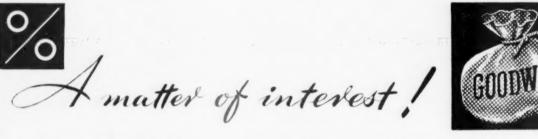
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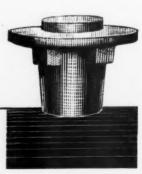


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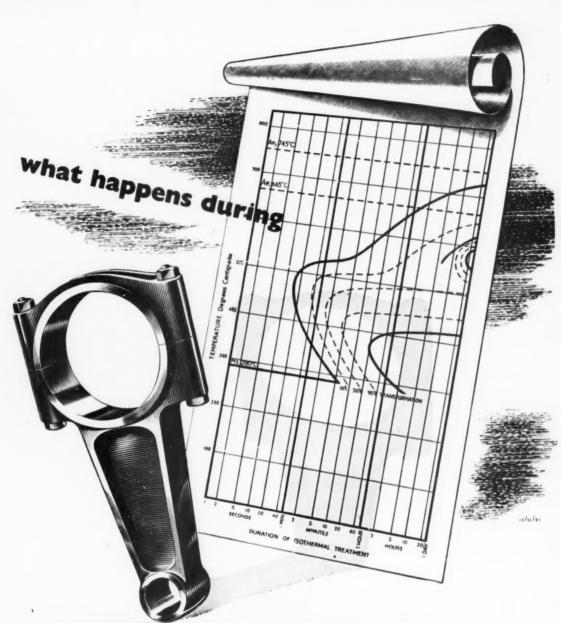
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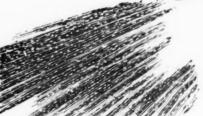




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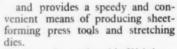
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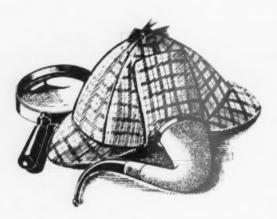
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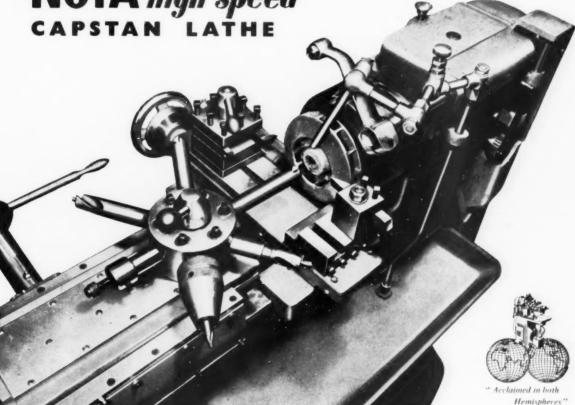
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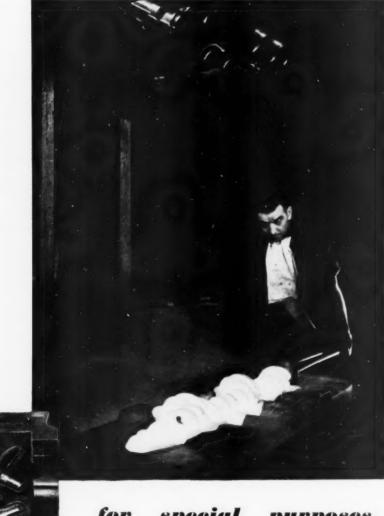


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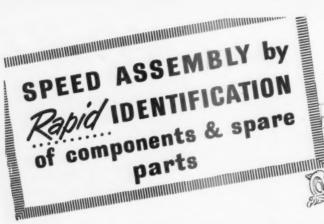
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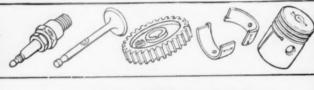


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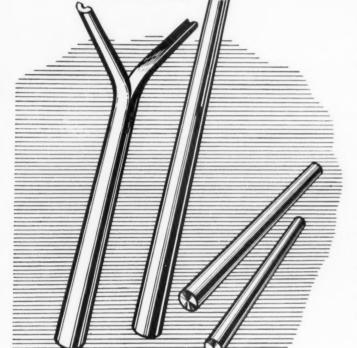
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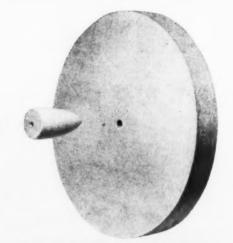
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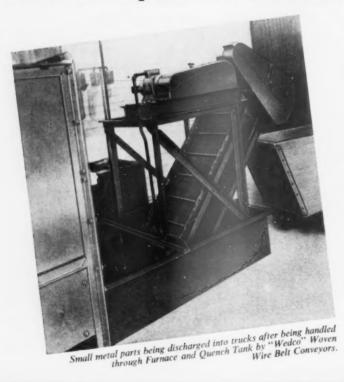
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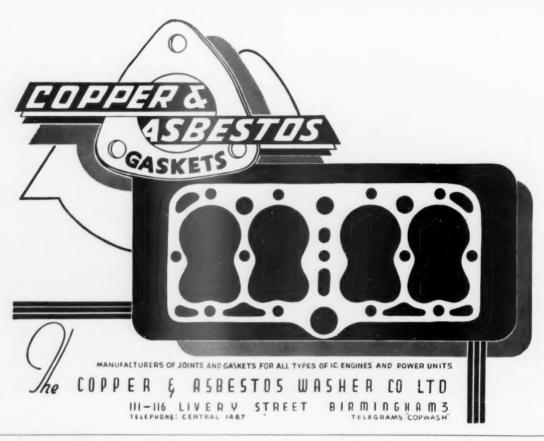


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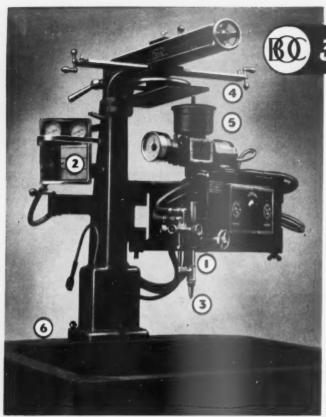




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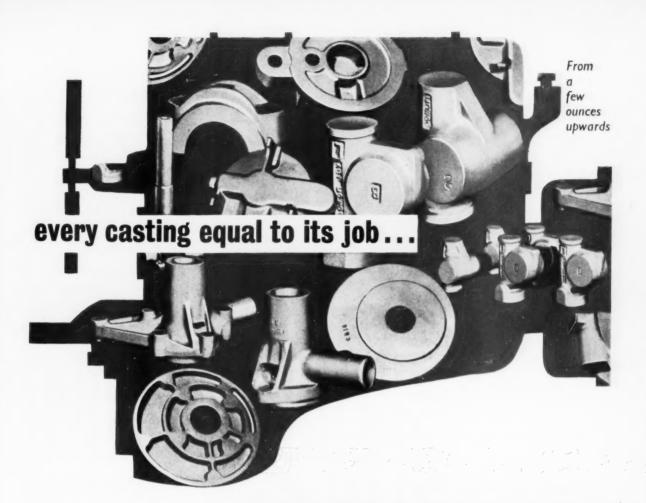
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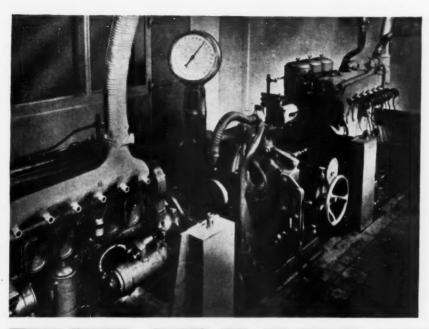
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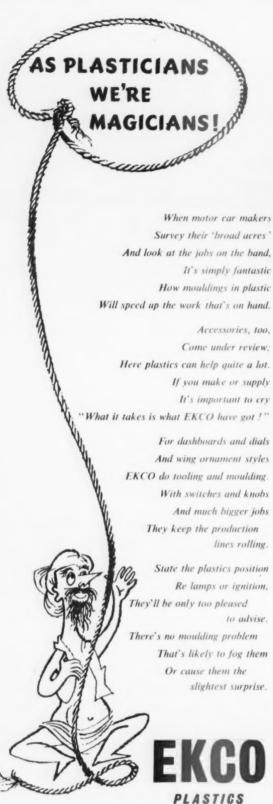
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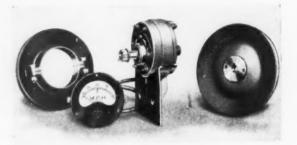
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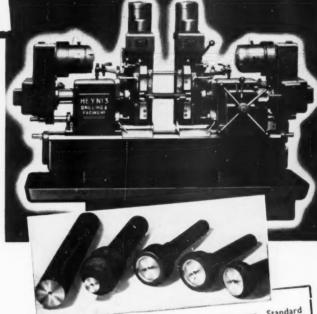
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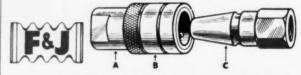
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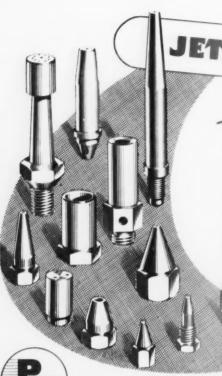
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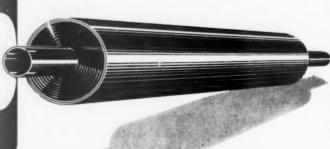
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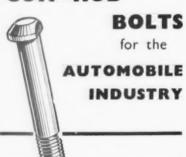
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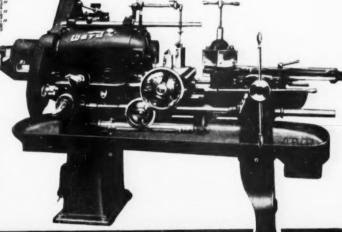
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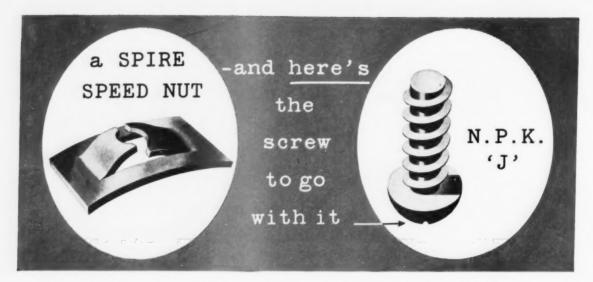
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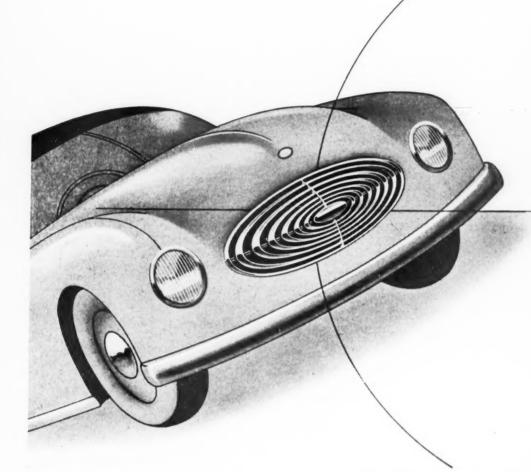
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